THE SLIDE VALVE
PRACTICALLY CONSIDERED.

N. P. BURGH.

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BY

N. P. BURGH, Engineer.

EIGHTH EDITION.

LONDON:
E. & F. N. SPON, 46, CHARING CROSS.
1878.

LONDON:
PRINTED BY WILLIAM CLOWES AND SONS,
STAMFORD STREET AND CHARING CROSS.

PREFACE.

To make this work better than the First Edition, I have rewritten it, and extended the matter in the following order:

Chapter I. contains the proportions for single ported slide valves, which I have treated at some length, explaining not only their use at the best, but also what to avoid. The formulæ are put forth in a simple and practical style for the purpose of general application.

Following on from this, I have written Chapter II. under the heading of exhaust relief, and double and treble ported slide valves; for these I have also investigated the proportions under

all circumstances, noticing in particular the width of the supply opening caused by the valve on the cylinder facing, width of the large bar, and amount of opening for the main exhaust port. The examples described in these chapters are taken from actual construction; the proportions therefore form a guide for future practice.

As the mechanical matters that relate to the outside lap of the slide valve *must* be noticed before producing that proportion, I have in Chapter III. considered the following questions:

The variation in the speed of the piston and crank pin; relation of the travel of the valve to the eccentric circle; and delineation of the paths of the crank pin and centre of eccentric.

I have endeavoured to explain their meaning as much as possible from the result of practice, and I think the subject will be found to have been well ventilated.

In Chapter IV. the geometrical demonstrations to produce the outside lap of the slide valve for

any point of cut-off, &c., have been fully explained.

Of foreign authors, Dr. Zeuner, a German, and Messrs. Long and Buel, Americans, have been referred to and quoted. The English authorities I have cited are Professor Rankine and Messrs. Watt. After that I have dealt with the matter, investigating and explaining the actual meaning and practical value of the *versed sines of the crank and eccentric arcs*, their application and reference, and the reason why the length of the eccentric rod *must* bear a *distinct* relation to the length of the main connecting-rod, and the position of the latter to that of the slide valve.

The application of the slide valve as an expansion valve has been explained in Chapter V.

Chapter VI. is an explanation of the proportions of modern slide valves in actual practice by the firms of Messrs. Penn, Maudslay, Rennie, Ravenhill, Watt, Napier, Dudgeon, Winter, Spencer, &c.

Single, double, and treble ported slide valves are described; also valves for compound engines and expansion slide valves; making in all *eleven* examples, fully illustrated, and all the main dimensions given.

In Chapter VII. the most modern types of packing rings and their means of adjustment for slide valves are explained and completely illustrated.

As a conclusion, Chapter VIII. treats of general observations, taking up certain matters and disposing of them as far as practice will admit.

The number of illustrations in the First Edition was only eighteen; this Edition has *thirty-eight*, with thirty-seven pages of additional descriptive matter; and thus the entire subject has not only been extended in explanation, but in illustration also, up to the practice of this date.

N. P. Burgh.

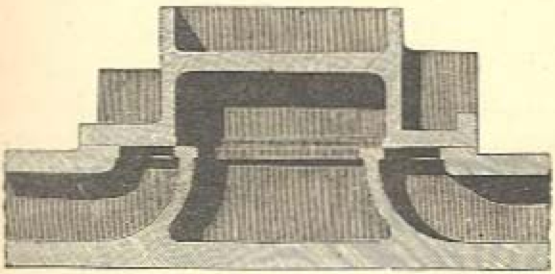
78, Waterloo Bridge, London, S.,
Sept. 1, 1868.

THE SLIDE VALVE.

CHAPTER I.

SINGLE PORTED COMMON AND EXHAUST RELIEF SLIDE VALVES.

SLIDE valves of the present day are divided into three classes or kinds—common, exhaust relief, and equilibrium. We therefore commence with the description of the common slide valve, which is illustrated by Fig. 1. This will be

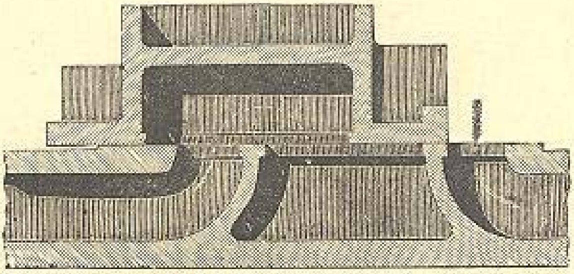


Common Slide Valve at half Stroke.

FIG. 1.

B

readily understood by the diagram, as to its shape, &c.; but it will not be out of place, perhaps, to give an example, as a reference for the proportions of the valve in relation to the ports in the cylinder. Let it be presumed, then, that a valve is required for an engine of 65-horse power nominal, adapted for the screw propeller. Allowing 17·5 square inches per horse power, the cylinder would equal about 38in. in diameter, having a stroke of 2ft. The area of the steam port supply being generally about 1 square inch per horse power, the result would be 65 square inches. Thus far having determined this proportion, it now remains to decide the proportion of the ports, &c., not at a guess, but by correct data, which will produce all the dimensions accordingly. It will be noticed next, in the dia-



Common Slide Valve at full Stroke.

FIG. 2.

gram, Fig. 2, that the valve, when at full stroke, has opened the port for the full width, hence the term "common slide." Now, it is obvious that the area of the port being given, and the same being divided by its length, the result will produce the requisite width, which latter dimension, in the present example, will be $2\frac{1}{2}$in., outside lap $1\frac{1}{2}$in., inside lap $\frac{1}{8}$in.; having thus far agreed, the following rules will supply what is required to complete the matter:

Width of cylinder bar = thickness of metal of exhaust passage × 1·25.

Width of exhaust space in valve = width of port supply × 1·5 + half travel of the valve + width of bar, minus inside lap.

Width of exhaust port in cylinder = width of bars minus inside laps, deducted from the width of exhaust space in the valve.

Now, as the two last rules may require a little explanation, it will be well to explain their origin. The exhaust space in the valve is that part which allows the steam to escape into the exhaust port. It is the general practice that the area of this port should exceed that for the supply; some engineers prefer supply 1, exhaust 2; others, the former 1, the latter 1·5. It is certain that when

the valve is at full stroke, as in Fig. 2, one bar only is in question as to the area for the exhaust; the half travel of any valve, it is almost needless to say, is the outside lap, plus the width of the supply port; thus it will be understood that the inside lap must be also taken into consideration to produce the exact proportion, although it may be said this last is not of vital importance, yet to attain the correct result, it must be noticed.

For explanation of the last rule, Fig. 1 will readily convey an idea of its origin and the truth of its composition; the inside laps, it will be seen, determine the dimension required. Simple as this rule may appear, it is of as much importance as those preceding it, more especially when requiring the width of the exhaust port by calculation in the absence of a diagram.

Returning to the requisite dimensions for the example given, they will be found to be thus:

Let A = width of port supply.
B = outside lap.
C = inside lap.
D = width of cylinder bar.
E = width of exhaust space in valve.
F = width of exhaust port in cylinder.
G = half travel of the valve.

	Inches.
Then, $A = \frac{65}{26} =$	2·5
$B = 2{\cdot}5 \times {\cdot}6 =$	1·5
$C =$	·125
$D =$	1·0
$E = 2{\cdot}5 \times 1{\cdot}5 + 1{\cdot}5 + 2{\cdot}5 + [1 - {\cdot}125] =$	8·625
$F = 8{\cdot}625 - [1{\cdot} - {\cdot}125 \times 2] =$ . . .	6·875
$G = 2{\cdot}5 + 1{\cdot}5 =$	4·0

The above calculations, and the results, represent the proportions of the valve and ports in Figs. 1 and 2. It may, perhaps, be well to add, that the formula given will be correct for any proportion, whether the valve is intended to cut off at one-quarter, three-eighths, one-half, five-eighths, three-fourths, or seven-eighths of the stroke. Also the exhaust may bear any proportion to the supply, remembering, of course, to use the ratio required, rather than the one given at present.

In Fig. 2, the valve, as before stated, is at full stroke. It will be seen also that the exhaust end of the valve overlaps or travels beyond the supply port. This overlap will be found, in all cases, to equal the outside lap, minus that of the inside. Therefore, when the valve has closed the supply port, the exhaust port will be open in

width, equal to the outside lap minus inside lap.

It need scarcely be added, that when no inside lap is given, the outside lap regulates the excess of time beyond that for the supply for exhausting. In the example now alluded to, the areas of the supply ports both for exhaust and supply, are equal; due, of course, to the valve travelling the entire width of the port. Now, to lessen the stroke of the valve would be to reduce the area for the admission of the steam, while that for the exhaust would not be affected, *i.e.*, should the reduction of the half-stroke of the valve not exceed the overlap on the exhaust side in Fig. 2. To retain the area for the supply of steam—but increase that for the exhaust more in proportion than in the diagrams Figs. 1 and 2, but not to exceed or reduce the stroke of the valve—the width of all the ports must be increased in proportion, hence the origin of the "exhaust relief valve."

For the purpose of future comparison and illustration, it will be better to give the proportions and formula for this valve. Now, presuming an engine of the same power as the last example, and the area for the supply steam to be equal, also the stroke of the valve—but allowing the width of the port to exceed that of the open-

ing—caused by the stroke of valve—the following result is obtained, viz.:—The exhaust side of the piston will be less susceptible to the action of the steam, which is generally known as back pressure or "cushioning;" the explanation of this will be fully described hereafter. The following formulæ are for exhaust relief valves:

Width of port = width of supply steam opening caused by valve × 1·5.

Width of exhaust opening caused by the valve = width of port × 1·5.

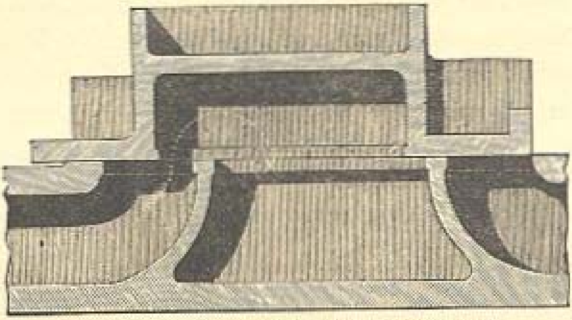
Width of exhaust space in valve = width of port × 1·5 + half travel of the valve + width of bar minus inside lap.

Width of exhaust port in cylinder = width of bars minus inside laps deducted from the exhaust space in valve.

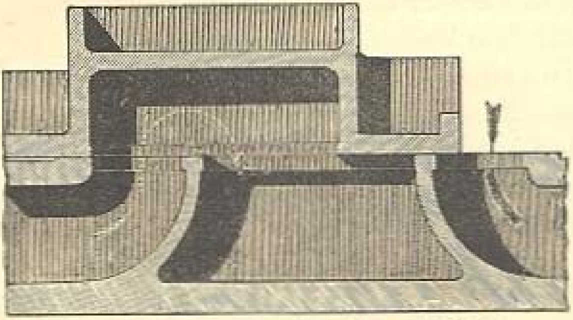
Before making use of these rules, it will not be out of place to explain their origin as for those previous. To begin with the rule for the width of the port, it will be seen that a constant number is given. Now, it must be strictly understood that this sum can be lessened or increased according to the discretion of the calculator, the number given being compiled from the average of general practice.

The second rule bears strict reference to the

space between the bar and the inside of the valve when the latter is at full stroke, hence the



Single ported exhaust relief Slide Valve cutting off.
FIG. 3.



Single ported exhaust relief Slide Valve at full Stroke.
FIG. 4.

increase of area or width of the exhaust port to that of the supply. The width of the exhaust space in the valve is produced by a formula, as in

the first example; also the last rule now given is the same in composition. In Fig. 3 will be seen a diagram of a valve and ports arranged for the exhaust relief principle. The valve is entirely covering one supply port, or cutting off, while that at the exhaust side is partially open; due, of course, to the arrangement before alluded to. In Fig. 4 the valve is shown at full stroke. Now, on comparing the widths of the ports for the exhaust in Figs. 2 and 4, the relative areas can be readily understood; the scale being the same throughout. It is, therefore, obvious that, while the valve is opening the port, for a given width as in Fig. 4, the width for the exhaust is considerably more, the increase being due to the proportions. Having thus briefly described the action of the valves Figs. 3 and 4; for the purpose of practically proving the same, attention will now be given to the calculations. As before stated, this valve is for an engine of the same nominal horse power as in the previous example.

Then, let—

A = width of steam supply opening caused by the valve.

B = width of port supply.

C = outside lap.

D = inside lap.

E = width of cylinder bar.
F = width of exhaust space in valve.
G = width of exhaust port in cylinder.
H = half travel of the valve.

	Inches
Let $A = \frac{65}{26} =$	2·5
$B = 2·5 \times 1·5 =$	3.75
$C = 2·5 \times \ ·6 =$	1·5
$D =$	·125
$E =$	1·0
$F = 3·75 \times 1·5 + 2·5 + [1· — ·125] =$	10·5
$G = 10·5 — [1· — ·125 \times 2] =$	8·75
$H = \ 2·5 + 1·5 =$	4·0

On referring to Fig. 4, the relative value of the above dimensions can be better understood. The supply port is opened $2\frac{1}{2}$in., but the port on the opposite side is opened full or $3\frac{3}{4}$in., and the distance from the inside of the bar to that of the valve will be $5\frac{5}{8}$in., or the width for finally exhausting; thus it will be understood the ratio of the ports to each other, is as 1 to 1·5. Now this proportion is not imperative, as 1 to 2, or 1 to $2\frac{1}{2}$, or 1 to 3, may be adopted, or, as in Figs. 1 and 2, they may be equal as far as regards the ports and openings.

It will now be well to allude to the lead on the

exhaust side. It should be understood that the only means of creating a perfect indicator diagram—the best mode yet known to indicate the action of the steam—is to cut off suddenly and exhaust freely. Should the latter not be accomplished the back pressure or extreme cushioning, before alluded to, will impede the action of the engine. It is, of course, well known that a lead on the steam side is often given, which will, to a certain extent, produce a cushion, or a round corner, in the indicator diagram; but when this is intentional, the evil may be said to be admissible. It will be seen, in Fig. 3, the lead of the exhaust is equal to the outside lap of the valve minus the inside lap. The same applies to Figs. 1 and 2, although not represented. It is, therefore, evident that the exhaust lead in the present rules is due to the outside lap minus that of the inside.

The adoption of the narrow bars is correct both in practice and theory, the friction being reduced in proportion to the wider bars. The larger area also is maintained for exhausting the steam freely for a given length of valve, or stroke of the same. In Fig. 1 the valve is represented at half stroke, and it can be readily understood that the valve has to move the remainder of the width of the

bar—on the exhaust side—before the area of the exhaust port is reduced; therefore, the steam can fairly exhaust at its greatest pressure through the larger area, both pressure and area being reduced proportionately at the same time.

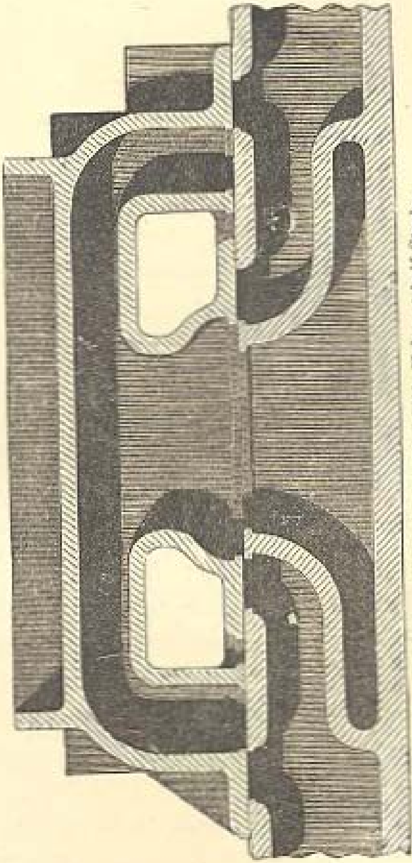
With reference to the inside lap of the slide valve, its vital intention is to ensure that the ports are covered on the exhaust side, so that no leakage of steam can occur when the valve is at half stroke.

CHAPTER II.

EXHAUST RELIEF DOUBLE AND TREBLE PORTED SLIDE VALVES.

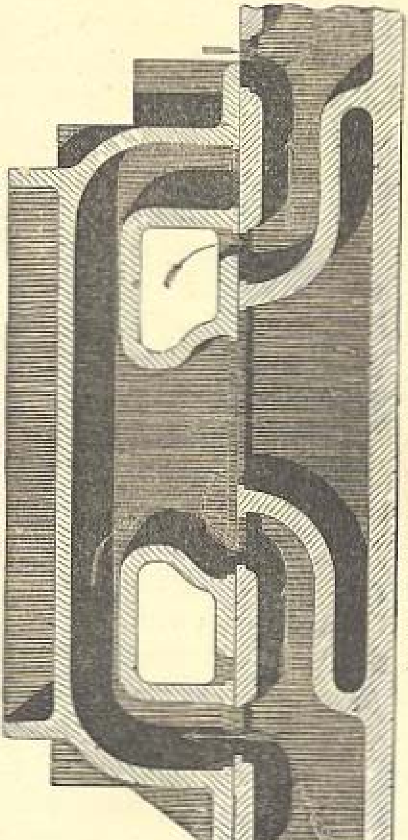
The common short slide valve, as previously described, has been long in use for engines of all powers and classes, but the friction with its use has greatly tended to retard its general adoption —hence the introduction of the double ported valve, commonly known as the "equilibrium or gridiron" slide valve. The term equilibrium is due to the peculiar shape of the body part of the valve, the steam thereby being allowed to act on the external and internal portions at the same time, equally or unequally, as the form will admit.

This arrangement is not entirely novel. It will be remembered, doubtless, by many, that the slide valves used for the old table and other engines were hollow between the faces; the supply steam



Double ported exhaust relief Slide Valve at half Stroke.

FIG. 5.



Double ported exhaust relief Slide Valve at full Stroke.

Fig. 6.

acting on the exterior portion, and the exhaust on the interior, thus causing an equalisation of the action of the steam, due to that portion it surrounded. The equilibrium slide valves of the present day are not much varied in principle, the difference being due rather to proportion than design.

Fig. 5, on page 14, is a good example, properly proportioned, both for supply and exhaust relief. It will be readily understood that the stroke of this valve will be exactly half of that of the common slide; due, of course, to the double ports in the place of single. Another great advantage in the reduction of the stroke of the valve for the same power is, in the proportions of the eccentric rods and links in a given space, particularly where direct action is required. The frictional surface is, perhaps, increased in the present example as compared with that of the common kind; but this evil is counteracted by the reduction of the stroke, and the almost neutral action of the steam on the valve. The valve seen in Fig. 5 is at half stroke or covering all the ports. Fig. 6, on page 15, will be understood to illustrate this valve at full stroke; the supply steam is indicated by full line arrows, while that for the exhaust is represented by the dotted kind, as in the preceding diagrams.

The often practical mode to ascertain the dimensions for a given valve has been by diagrams on slips of wood, cutting and carving them, as the case requires. Now, presume the absence of diagrams and the facility for producing any, calculations *properly founded*, will produce a correct result, without fear of doubt as to the practical value of the dimensions attained. In order to prove the truth of this statement, presume that the diagrams on pages 14 and 15 illustrate a valve and ports for a cylinder of 50-horse power nominal. Let it be assumed that the width of the opening caused by the slide is 1in., the outside lap to be $1\frac{1}{2}$in., the inside lap to be $\frac{1}{16}$in., and the width of the narrow, or inside, cylinder and valve bars, to be $1\frac{1}{4}$ in. each. Having decided on these dimensions, the remaining proportions are produced from formulæ. Now, for engines moving at high velocities, it is necessary that the exhaust should have a lead on the supply, which will, of course, as before stated, be due to the width of the outside lap, minus inside lap. The proportion of the ports supply to that for the openings is not imperative. The same may be said of the exhaust port in relation to that for the supply. Presuming the width of the latter to be 2in. each, the following formulæ will supply the requisite dimensions.

Width of exhaust space in valve = width of two ports supply × 1·5 + half travel of the valve + width of small cylinder-bar minus inside lap.

Width of exhaust port in cylinder = width of small cylinder-bars, minus inside laps deducted from the width of exhaust space in the valve.

Width of large bar in cylinder = outside lap + width of opening caused by valve + width of small bar in the valve + half travel of valve.

Or, the formula can be arranged thus: travel of valve + width of small bar in the valve. When the width of the *small* port in the slide valve exceeds the width of the opening caused by the slide, this rule must be: *travel of valve + width of small bar + the excess alluded to.*

To test the truth of these rules will be to disarrange all the proportions, but before entering into that subject it will be well to explain the reason for the present constants. The "width of the exhaust space in the valve"—on referring to Fig. 5, in page 14—will be understood to be based on two fixed proportions, viz. ratio of the supply ports to that for the exhaust, and the half travel of the valve. On referring to Fig. 6, in page 15, it will be seen that one bar only affects the exhaust from the outside port, hence the allusion in the rule to one bar. The

last portion of this rule may seem superfluous, but when a certain dimension, or rather proportion, is fixed upon, the inside lap must be considered. Another reason for its introduction is, that the valve, when at full stroke, partially covers the centre or exhaust port in the cylinder—see Fig. 6. It is, therefore, requisite to observe the whole of the formula given to attain a correct or fixed proportion of supply to exhaust.

The next rule, "width of exhaust port in the cylinder," is based on the result of the first formula. On referring to Fig. 5 this will be apparent, and the relationship readily understood. It will be better to add that narrow bars are not imperative, although now generally introduced; and that were wide bars used, the formula under notice would be the same.

The third rule is the "width of the large bar." This dimension is the most important of all, as from or on it must be set out the openings, or ports and bars, in the valve. Now, to some this undoubtedly may seem a simple process; but in the absence of a diagram a correct formula is invaluable.

To return to Fig. 5 again will perhaps assist the mind more readily in understanding the principles on which this formula is based. Let it be

presumed that the valve is removed from the face of the cylinder, the plain surface will be, perhaps, perplexing, and the reasons for the fixed dimensions mysterious. On glancing at Fig. 6 the truth will be apparent. For example, the large bar in the present illustrations is $6\frac{1}{4}$ in. deduced by the formula. In Fig. 5 the valve is at half stroke. Let it be presumed that the act of testing the truth of the dimension is in operation. Commencing on the inside part of the large cylinder bar, we set off on it towards the outer side, or from the centre line, the outside lap. From this last point is set out the width of opening caused by the valve, or the width of the supply port in it; next the width of the small valve-bar. From this point set off the half travel of the valve, and the result will be the dimension alluded to. Thus far, having proved the basis of the formula for a given dimension and proportions, it may with justice be argued that, as only one example has been introduced, further explanation is requisite. To further demonstrate by different proportions would at present be confusing, until those alluded to are disposed of.

Then, let—

A = width of steam supply opening caused by the valve.

B = outside lap of valve.
C = inside lap of valve.
D = width of port supply.
E = narrow bar.
F = half travel of the valve.
G = width of exhaust space in the valve.
H = with of exhaust port in the cylinder.
I = width of large bar in the cylinder.

	Inches.
Then A = ...	1
B = ...	1·5
C = ...	·0625
D = ...	2·0
E = ...	1·25
F = 1 + 1·5 = ...	2·5
G = 2 + 2 × 1·5 + 2·5 + [1·25 − ·0625]	= 9·6875
H = 9·6875 − [1·25 − ·0625 × 2]	= 7·3125
I = 1·5 + 1 + 1·25 + 2·5	= 6·25

These calculations and their results bear strict reference to Figs. 5 and 6. It will be seen in Fig. 5 that the exhaust passage in the valve is wider than the port supply, and also overlaps the same. Now, this proportion of passage to port is partially due to the proportion of the half travel to the port. It will be well to add that the correct overlap can be known by F − [D + C].

As before stated, the proportion of the large bar is the most important; therefore further remarks will not be out of place. The overlap on the large bar will, of course, be equal to that already alluded to. The constant number given in the formula for the large bar is not a fixed sum; also any proportion of lap to the width of the port or opening, to cut off at a given stroke of the piston, can be maintained; but it is almost needless to add that the overlap will be increased in proportion also. For example, let it be presumed the values of

	Inches.
A = ...	1
B = ...	3
C = ...	0·125
D = ...	2·0
F = ...	4·0

Then the overlap = F − [D + C] = 1·875, increasing the large bar in correct proportion also.

Now, as the two examples given are in proportion to each other, *i. e.*, with an excess of overlap, and lead for the exhaust; also the large bar exceeds the total of A, B, C, D, and E; due, of course, to the proportions decided on. We will next imagine a valve to be proportioned so that there shall be no overlap—but this is given for

comparison only, not as a correct proportion. In order to make this more concise as well as practicable in all cases, the following formula will greatly assist. Let F = D + C, commencing then with this formula, let

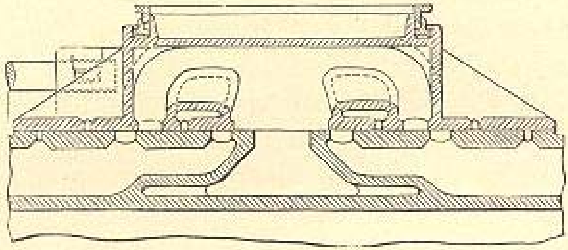
	Inches.
A = ...	1
B = ...	1·0625
C = ...	0·0625
D = ...	2·0
E = ...	1·25

Then I = A + B + C + D + E = 5·375.
Again, I = B + A + E + F = 5·375.

By this it will be seen that the results are alike, thus proving the truth of the concise formula alluded to. It is well known that, in some cases, for practical reasons, the laps and leads of the valves now under notice are unequal, in order to preserve a correct action. To attain this the width of the steam supply and exhaust ports in the valves, as well as the large bars in the cylinder, would be varied, but the formulæ here given would not be materially affected.

From the foregoing conclusions it is evident that the width of the steam opening caused by the valve has much to do with the travel of the

same, and that to reduce the travel, the width must also be contracted. It has been shown that by having two ports in the place of one, this can be accomplished in proportion; and we now follow on to a still greater reduction of travel by the use of the *three* ported valve, as shown by Fig. 7.



Treble ported exhaust relief Slide Valve at half Stroke.

Fig. 7.

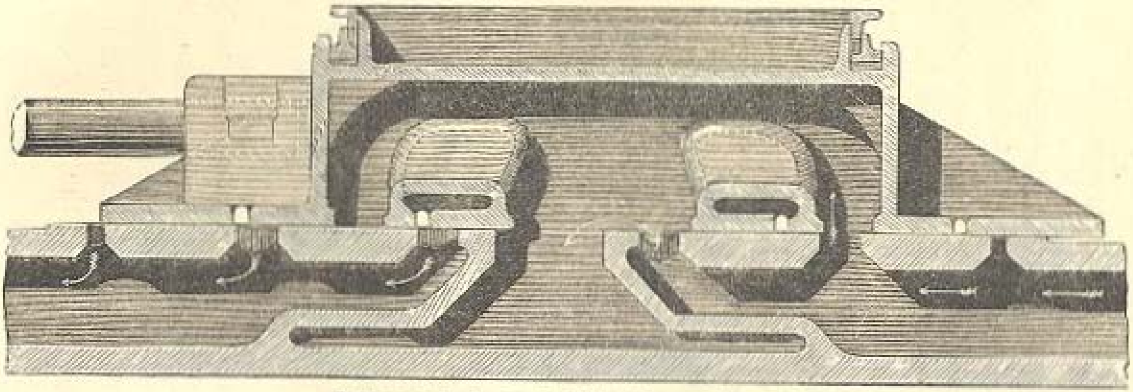
Now, if we return to Fig. 5, in page 14, we shall see that the cylinder has *two* ports on each side of the small bars confining the main or central exhaust port. On comparing that with the present example, it will be seen that on each side of the small bars there are *three* ports, two of which are wider than the third, and that there are also two wide bars in the place of one, as in Fig. 5.

We have fully demonstrated the fact that the

width of the large cylinder-bar, under ordinary circumstances, to be of proper proportion, should equal the travel of the valve plus the small bar in the valve; and we may now add that this rule is really the basis of the method for producing the cylinder bars in the present illustration, for it is plain that the width of each large bar here also equals: the outside lap + the width of steam port in the valve + the small bar in the valve + the half travel of the valve. As there are two wide bars in the cylinder on each side of the small bars, it is essential that the widths of these should be equal, and thus each are produced from the above formula.

It may be noticed next that the two inside wide bars on each side of the exhaust ports in the valve are both of the same width, and that they are produced by this formula; inside lap + width of one small exhaust port in the cylinder + outside lap. The two widest bars in the valve at the extremities are equal in width to the large bars in the cylinders.

To enable this matter to be more easily understood, the illustration, Fig. 8, on the next page, is introduced at a larger scale than that of Fig. 7. The valve is shown at full stroke, and the arrows indicate the passing of the supply and exhaust



Treble ported exhaust relief Slide Valve at full Stroke.

FIG. 8

steam; where it is shown that the supply steam is admitted through *three* narrow ports, and the exhaust steam out through two wider ports. The main proportions of valves of this class are produced from the following formulæ:

Width of one supply steam opening in the cylinder caused by the slide valve = one-third of the total width of the area required.

Width of one wide port in the cylinder = width of one supply opening × 3 or 4, as may be agreed on, to allow a full area for the exhaust steam.

Width of main exhaust opening in the cylinder caused by the valve = width one cylinder port × 2.

Width of main exhaust port in the cylinder = width of main exhaust opening + half travel of valve - [width of small bar in cylinder - inside lap of valve].

Width of main exhaust space in the valve = width of main exhaust port in the cylinder + [width of both small bars in the cylinder - inside laps].

Width of large bar in the cylinder = travel of valve + width of small bar in the valve.*

Width of large bar in the valve = outside lap + width of small exhaust port in the cylinder + inside lap.

* Also see formula in page 18.

Width of outside large bar in the valve = width of large bar in the cylinder.

Width of end exhaust port in the valve = width of large bar in the cylinder – [width of small bar in valve + width of small port in valve + outside lap + inside lap].

Width of small ports in the valve and cylinder = width of one steam opening caused by the valve.

To prove the veracity of these formulæ, we must put them in order as in the preceding examples, and, as the best means of doing so, we will refer to a valve of this class designed by the author for one of the cylinders of a pair of marine engines of 800-horse power collectively nominal:

A = width of steam supply opening caused by the valve.

B = width of small exhaust port in the cylinder.

C = outside lap of the valve.

D = inside lap of the valve.

E = half travel of the valve.

F = width of exhaust opening caused by the valve in the cylinder.

G = width of narrow or small bar in the cylinder.

H = width of main exhaust port in the cylinder.

I = width of main exhaust space in the valve.
J = width of large bar in the cylinder.
K = width of large bar in the valve.
L = width of outside large bar in the valve.
M = width of narrow bar in the valve.
N = width of end exhaust ports in the valve.
O = width of small ports in the valve and cylinder.

Then, as—

	Inches.
A =	1·375
B =	4·0
C =	3·25
D =	·0625
E = 1·375 + 3·25 =	4·625
F = 4 × 2 =	8·0
G =	1·25
H = 8 + 4·625 − [1·25—·0625] =	11·4375
I = 11·4375 + [1·25 × 2 − 2 × ·0625] =	13·8125
J = 4·625 × 2 + 1 =	10·25
K = 3·25 + 4 + ·0625 =	7·3125
L =	10·25
M =	1·0
N = 10·25—[1 + 1·375 + 3·25 + ·0625 =	4·5625
O =	1·375

We have thus given the face dimensions of the valve and cylinder as far as the widths of the

ports and bars are concerned, and we may now add that the formulæ are applicable for any proportions. The main matter, indeed, is but questions of laps, widths of openings and ports; therefore the remainder follows in direct proportion, unless an unequal dimension at any part is agreed on. It is as easy to proportion a valve for four or six openings as two, by remembering the *relation* of the lap to the opening, which will be explained in detail in the next chapter, the present being merely the arrangement of the formulæ when the main features of the proportions are agreed on, together with a description of slide valves of the most popular kind.

CHAPTER III.

THE MECHANICAL MATTERS THAT RELATE TO THE OUTSIDE LAP OF THE SLIDE VALVE.

The variation in the speed of the piston and the crank pin.—To put this matter in the most practical and correct form, the diagram, Fig. 9—requisitely put in pages 40 and 41—is introduced, which represents the centre lines of a connecting-rod and crank at *five* positions. The stroke of the piston is 3 ft. 6 in., and the length of the connecting-rod between centres is 9 ft. 6 in. The plane, or horizontal length of the motion, is divided into six parts, and the points of cut-off are five. As the motion of the piston and that for the connecting-rod pin are alike, it can be readily understood that the points 1, 2, 3, 4, and 5, on the plane line, refer to the piston also; and the points on the crank circle apply in strict relation to the crank pin.

Knowing this, and knowing also that the piston-

rod connects the end of the connecting-rod to the piston, the latter can be ignored in the description, and the lineal motion of the connecting-rod only alluded to.

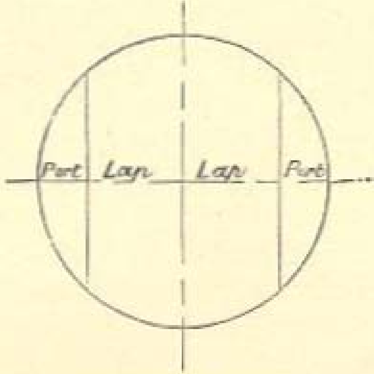
Suppose, now, that the connecting-rod is at the inner end of its path, the crank will be on the horizontal line of the circle. The end of the rod is next assumed to be at the point 1, and has, therefore, travelled back 7 in.; but the crank pin has moved on its path a length of about three times that, while the dotted arc of the crank's position shows the same advance as the connecting-rod pin. The latter then moves to No. 2 point, and also the crank pin on its circle to the same number of position. In this case the difference in the lengths of the travel of each pin is considerably reduced, being in the proportion of about as 7 is to 9 only. This becomes less in relation to the travel of each pin during from 2 to 3 and from 3 to 4; while from 4 to 5 it is nearly equal to the proportion of that from 1 to 2, and from 5 to the end of the stroke the ratio is about 7 to 12·35. It will, of course, have been noticed that the lineal positions of each pin on the horizontal line are equal throughout, those for the crank pin being indicated by the dotted arcs within the circle.

From this diagram, or any other of the same principle, the difference in the speeds of the crank pin and piston can be faithfully depicted during the several portions of the motion.

For example, if the piston only moved 7 in. from the end of its path to the point 1, while the crank pin moved 21 in. on the length of the arc of the circular motion, the speed of the piston *must* be one-third of that of the crank pin; so that by knowing the *ratio* of the *distances* passed through by the two pins in the *same time*, the difference in the speed of each is easily obtained.

Relation of the travel of the valve to the eccentric circle.—When the length of the motion for the slide valve is the same as the diameter of the eccentric circle, the action is termed "direct," which, indeed, is the present practice as a rule with few exceptions. The circle referred to is really the same in relation to the slide valve's motion as the crank circle is to the path of the piston, which we have just explained.

To enable the present subject to be understood at once, without wading through much description, we introduce the diagram, Fig. 10, on the next page, which shows the path of an eccentric. Let it be assumed it is 6 in. in diameter, the width of opening—termed port in the diagram—is 1 in.,



Scale of 3 in.=1 foot.
Diagram of the Path of an Eccentric.
FIG. 10.

and the remainder of the half diameter will be the outside lap, 2 in.; the vertical lines depict the points of division, so that it is obvious that the lineal width of the eccentric circle is divided into four parts, thus: ports or openings opposite each other, at the extremities, and the laps between.

Delineation of the paths of the crank pin, and centre of eccentric.—It is doubtless universally known that, virtually, the crank path is divided into four distinct parts, also that for the eccentric. The proportions of these divisions are practically regulated by the grade of expansion agreed on to be maintained. Fig. 11 represents a crank path, the chords

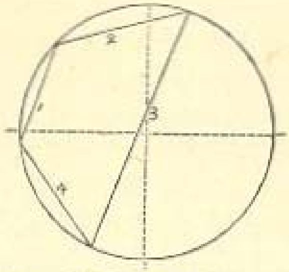


Diagram of the Path of the Crank Pin.

FIG. 11.

indicating the divisional points, the relative proportion is, therefore, readily understood. No. 1 is the chord of supply, No. 2 that for expansion, No. 3 relates to exhaustion, and No. 4 represents neutrality, or that portion of the stroke of the piston when the port on the exhaust side is covered, often termed compression, consequently, the piston for a period is devoid of pressure or vacuum. The length of the chord 1 is due to two causes, the grade of expansion and the length of the connecting-rod. It will be noticed that the chord at the plane line intersects with the circle slightly below the same, this last intersection is the angle that the crank assumes when the slide valve commences to open the port, and the vertical distance from the intersection to the plane line is due to the lead re-

quired. The upper point of intersection, as before explained, is subject to the curve assumed by the connecting-rod from a point, or distance, on the plan line to the circle of the crank path. The length of chord No. 2 is regulated by the inside and outside laps of the slide valve. The expansion of the steam is now presumed to be in full operation, and is released by the opening of the port on the exhaust side, hence the intersection of the chord at No. 3. The length of the last-mentioned chord is, more than any other, due to the traverse of the valve, or the time occupied in opening and closing the port for exhaustion. Now in the case of an increase of supply steam, the time for expansion and exhaustion would be lessened in proportion, it being remembered that the circle described by the crank pin cannot be increased or decreased for a given length of stroke of piston. The circle, as before stated, is divided into four divisions, and the alteration in the grade of expansions or length of connecting-rod affects the whole proportionately. The concluding chord, No. 4, represents neutrality, or, as before stated, that portion of the stroke of the piston where the vacuum and steam is cut off for a given period, commonly known as compression. It will be remembered that the chord of expansion is due to

the outside and inside laps, *i.e.*, when the limit of the valve is at the edge of the supply side of the port, the valve has to travel forth until the inside edge permits exhaustion or destroys expansion—see Fig. 3, in page 8; directly this ensues the value is at half stroke, plus inside lap; exhaustion then continues until the valve has resumed its position, but travelling in a reverse direction. The position of the valve when terminating exhaustion will be half stroke, minus inside lap. It is shown that the length of the chord for expansion and compression, are nearly equal in the example given; it may also be added that any variation in these two chords will depend on various causes, such as unequal laps and leads, &c. &c., but with certain arrangements both are equal. The fact of the compression being the same as the expansion, is of no vital importance; it is certain there would be a gain in maintaining expansion longer, and exhausting, till the supply commenced, and thus dispensing with compression; but the present mode of motion for the slide valve would have to be discarded, as an extreme varied movement is essential for the purpose.

The proportions of the path of the crank pin can thus be clearly understood. As at present

arranged—compression, supply, and expansion, from one portion of the circle, and the remainder is occupied during exhaustion, the latter function being the largest in operation and time.

It may be added—that the diagram under notice relates to one stroke of the piston only, and therefore to only half a revolution of the crank pin; there is, however, a slight variation in the lengths of the chords on the return stroke, due especially to the angle of the eccentric in relation to that of the crank. *Obviously, then, the arrangement of the slide valve, in relation to that of the connecting-rod, should be considered as to the attainment of equal action; for the versed sines, of the arcs of the chords, passed through by the crank pin, for such angle of cut-off for a given supply of steam are unequal, also that for the eccentric, when the valve is at the same side of the crank as the connecting-rod.* This, however, can be counteracted to a certain extent by short eccentric rods, and levers, reverse in action, and situated to compensate for the inequality of the speed of the circular and sliding motions, the attainment of which is of great importance.

As an illustration of this, Fig. 12 represents the circle described by the crank pin for a given

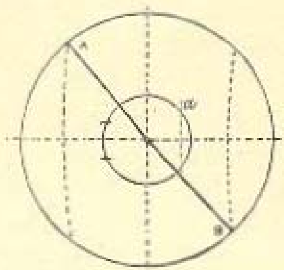


Diagram of the Angles of the Crank and Eccentric.

FIG. 12.

stroke of piston. The horizontal or plane line is presumed to be the centre line of the motion, and the larger dotted arcs represent an even grade of expansion, on each side of the piston or at each stroke. Now it will be seen that on the crank — represented by the thick lines — moving from the plane to the intersection at A, a given length of stroke of piston is produced, due, of course, to the radius of the larger dotted arc. On the crank reaching to the intersection at B, the grade of expansion is reversed in action, but the same distance from the end of the stroke retained. Obviously, the difference in the lengths of the arcs of the circle passed through are due to the length of the dotted arcs, the radii of which are the connecting-rod. The smaller circle, *a*, denotes the travel of the valve or the path of

the centre of formation of the eccentric. The lesser dotted arc indicates the distance the valve must be from the edge of the port when the piston is at full stroke, hence the angle or advance of the eccentric to that of the crank

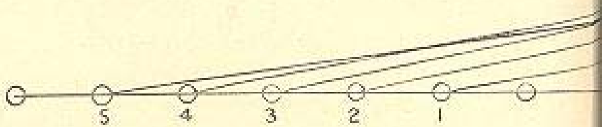
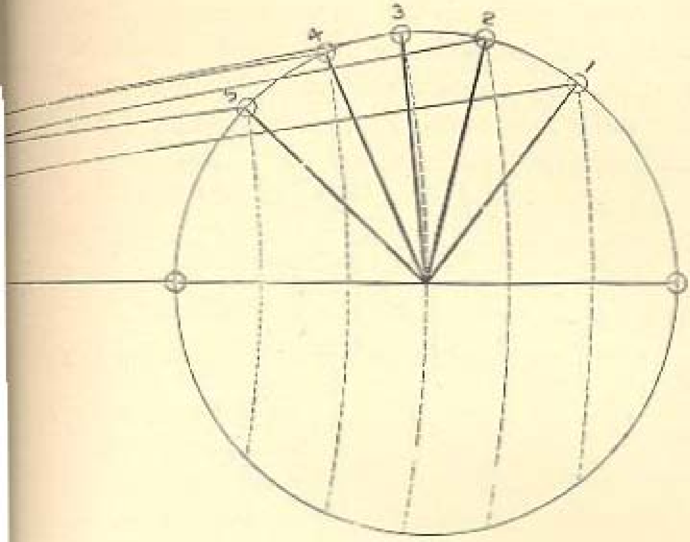


Diagram of the Paths of the Connecting-rod Pin and the Crank Pin.

Scale $\frac{3}{4}$ inch = 1 foot.

Fig. 9.

when on the horizontal line, or when starting to rise to A. When the crank is at A, the eccentric has passed through an arc equal in length proportionately. The two dots on the circle opposite the dotted arc indicate the angles of the eccentric when the crank is at the plane line and at B. Now it will be readily seen that the space between the dots is less than that of the intersections opposite; also the versed sine reduced: evidently, then, an unequal grade of expansion must occur on the side of the circle at B, *i.e.*, if the laps of the valve are equal. It must also be remembered that, to increase the lap and retain the previous stroke of the valve is



to destroy the lead; hence, the slight variation in the time, for the supply of steam and its expansion for opposite strokes of the piston; which may be endured rather than introduce a worse evil, or the loss of lead at one end of the cylinder.

There is no intimation here that perfection of mechanism should not be attained, but rather to explain what is the general practice at present. The diagram now under notice—not the one above, but that in the preceding page—illustrates the principle of the action of the crank and eccentric with the valve situated on the same side of

the crank shaft as the connecting-rod, direct action in each case being maintained. It will be seen that the arcs are all struck by the radii on the same side of the perpendicular line, hence the variation just described.

To counteract, or rather obviate, the imperfections in question, the versed sines of the chords indicating the partial stroke of the valve must be equal on each side of the perpendicular line; for example, Fig. 13 represents a crank pin path,

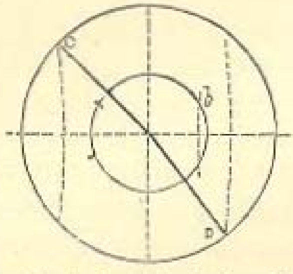


Diagram of the Angles of the Crank and Eccentric.

FIG. 13.

and that of the eccentric, to cut off at the same grade of expansion as Fig. 12. In the present case the arcs passed through are reversed to that of the former, the piston is presumed to be moving in the same direction, but the position of the connecting-rod opposite to that of the slide

valve. It will be noticed thatthe diameters of the eccentric paths are unequal; this inequality is due to the variation in the arc of the crank pin's passage during the forward and backward grades of expansion. When the crank has moved from the plane line to C, the eccentric has passed through an equal arc, and the same relative motion occurs from the horizontal line to D. Now the length of the intersection at *b* can be seen, in proportion to that between the dots in the circle opposite. Suffice it to say, the nearer these two intersections agree in length or space between the same, the less variation will ensue in the grades of expansion at each double stroke of the piston. It will be understood that the positions of the pistons in Figs. 11, 12, and 13 are presumed to be alike; also the direction of the movement. It may as well be added that on reversing the action of the piston the travel of the valves would be effected, and that in some instances the greater travel of the valve is preferred, for the purpose of obtaining an equal lead with an equal grade of expansion for each stroke of the piston.

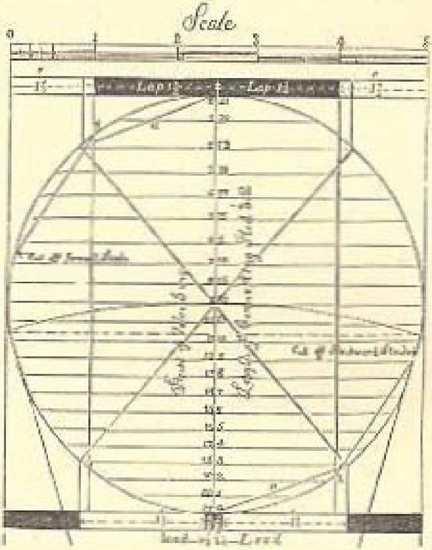
CHAPTER IV.

GEOMETRICAL DEMONSTRATIONS TO PRODUCE THE OUTSIDE LAP OF THE SLIDE VALVE, FOR ANY POINT OF CUT-OFF, ETC.

THE subject now entered on has been well digested by many authors, both English and foreign. Of the latter the best authority is Dr. Gustav Zeuner, a German gentleman: his researches and expositions pertain, however, more to the locomotive than the marine type; but, in the main, his views are correct for either, under relative circumstances. The views of our American friends are represented in a work termed the "Cadet Engineers," by Messrs. Long and Buel. At home, Messrs. Watt, Professor Rankine, Mr. D. K. Clarke, C.E., and Mr. M'F. Gray, claim attention for their productions, and the Author has also done his best to solve the problem.

Geometry of the Slide Valve, by Messrs. Watt.

Messrs. Watt's mode of treating the matter under notice is illustrated by Fig. 14, which repre-



Messrs. Watt's Mode of producing the Points of Cut-off by a known Lap

Fig. 14.

sents the path of a crank pin 21 in. in diameter; length of connecting-rod between centres, 3 feet 2 in., and the stroke of the slide valve, 5 in. The method of utilising the diagram is thus: "Divide the path of the crosshead pin into inches; with the

connecting-rod's length as a radius, and each inch as centres, describe arcs, intersecting with the path of the crank pin; join the intersections by chords or lines, as depicted, and the result is a correct illustration of the relative position or progress of the crank pin and piston. Next draw vertical lines parallel to the centre line as tangents, and above the circle, between these tangents, from a scale of as many equal divisions as the number of inches in the stroke of the valve, which is assuming, that virtually the stroke of the piston is that of the valve. Now, having previously settled the outside lap of the slide valve, next draw two tangents parallel with the centre *horizontal* line, and on the *top* tangent on each side of the *vertical* centre line, set off the "lap" according to the "scale:" the lap being 1½ in. actually, 1½ divisions in the scale is the length required, each main division in the scale being virtually inches. Draw vertical lines from the "laps" cutting the "circle," and prolong them to the *bottom* tangent. On each side of the centre line set off the "lead" of the slide valve, at the same scale as the "lap" on the lower tangent, and from the "lap" set off the "leads" also. From these last points draw lines parallel with the vertical centre line, to intersect with the "circle" above and below; connect these intersec-

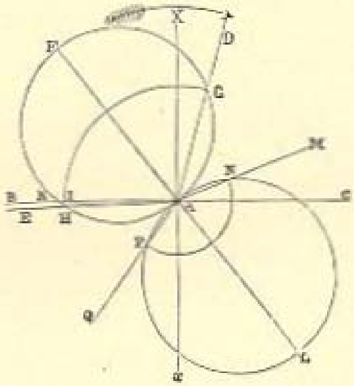
tions by angles crossing each other at the centre of the circle. Then with the chord *a* as a radius, and *b* as a centre, describe arcs on the circle, oppositely, and these last intersections are the centres of the crank pin, when the slide valve has closed the port supply, or points of cut-off, for the forward and backward strokes of the piston. The curve seen intersecting with the circle's centre is described by the connecting-rod when the piston is at half-stroke, and, by a similar process, the position of the piston can be ascertained at the points of "cut-off," which in this case is a mean of 12·82 in. from the commencement of the stroke. As the top and bottom "tangents" represent the steam and vacuum sides of the slide valve, the position of the latter, and angles of the eccentric, are also obvious.

The Geometry of the Slide Valve, by Dr. Zeuner.

Professor Rankine, when treating of the "Slide Valve Gearing," in his work of "Rules and Tables," states:

By the *angular advance* of the eccentric is to be understood the angle at which the eccentric radius stands in advance of that position which would bring the slide valve to mid-stroke when the crank is at its dead points.

RULE I.—Given, the positions of the crank at the instants of admission and cut off; to find the proper angular advance of the eccentric, and the proportion of the lap on the induction side to the half travel of the slide.*



Dr Zeuner's Geometrical Diagram of the Slide Valve and Crank.

FIG. 15.

In Fig. 15 let A B and A C be the positions of the crank at the beginning and end of the forward stroke; let the arrow show the direction of rotation; let X x be perpendicular to B C; let A D be the position of the crank at the instant of cut off, and A E its position at the instant of ad-

* The method used in this and the following rules is that of Professor Zeuner of the Swiss Federal Polytechnic School at Zürich, published in his treatise on Slide-valve Gearing, entitled *Die Schiebersteuerungen*.

mission. Draw A F, bisecting the angle E A D; A F will represent the position of the crank at the instant when the slide is at the *forward end* of its stroke; and F A X will be the *angular advance of the eccentric.*

Lay off the distance A F to represent the half travel; and on A F as a diameter describe the circle A H F G; cutting A D in G and A E in H; then $\frac{A\,G}{A\,F} = \frac{A\,H}{A\,F}$ will be the *required ratio of lap at the induction side to half travel*; and A G = A H will represent that lap, on the same scale on which A F represents the half travel.

On the same scale I K represents the *width of opening of the valve at the beginning of the stroke,* sometimes called the "*lead of the slide.*" Strictly speaking, this is the lead of the induction edge of the slide only; the lead of the centre of the slide being A K; that is, its distance from its middle position at the beginning of the forward stroke.

Rule II.—Given the data and results of the preceding rule, and the position A M, of the crank at the instant of release; to find the ratio of lap on the eduction side to half travel, and the position of the crank when cushioning begins. Produce F A to L, making A L = A F; on A L as a diameter

draw a circle, cutting A M in N: then $\frac{AN}{AL}$ will be the *required ratio of lap at eduction side to half travel.*

About A draw the circular arc N P, cutting the circle A L again in P; join A P; then A P will be the required *position of the crank at the instant when cushioning begins.*

Rule III.—Given, the data and results of Rule I., and the position, A Q, of the crank at the instant of cushioning, to find the ratio of lap at the eduction side to half travel, and the position of the crank at the instant of release—produce F A as before; on A L = F A as a diameter draw a circle cutting A Q in P: $\frac{AP}{AL}$ will be the *required ratio of lap at the eduction side of half travel.*

About A draw the circular arc PN, cutting the circle A L again in N; join A N: A N will be the position of the crank at the instant of release.

Rule IV.—Given, the angular advance of the eccentric, the half-travel of the slide, and the lap at both sides; to find the positions of the crank at the instants of admission, cut-off, release, and cushioning. Draw the straight lines B A C and

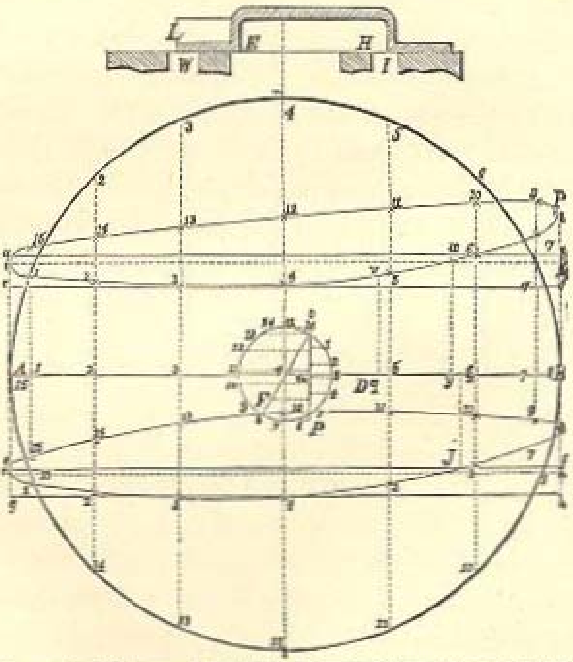
X A *x* perpendicular to each other; and take B and C to represent the dead points. Let the arrow denote the direction of rotation. Draw F A L, making the angle F A X = the angular advance of the eccentric; and make A F = A L = half-travel. On A F and A L as diameters, draw circles. About A, with a radius equal to the lap at the induction-side, draw an arc cutting the circle on A F in H and G; also, with a radius equal to the lap at the eduction-side, draw an arc cutting the circle on A L in N and P Draw the straight lines A H E, A G D, A N M, A P Q. These will represent respectively the positions of the crank at the instants of *admission*, *cut-off*, *release*, and *cushioning*.

The Geometry of the Slide Valve, by Messrs. Long and Buel.

The information alluded to in the "Cadet Engineer" is described and illustrated as follows:

Now, if we wish the port to be closed before the termination of the stroke, we make the face of the valve longer, or put on *lap*. In this case the throw of the valve must be increased by an amount equal to twice the lap. But if excessive lap be put on, it is evident that the travel will be

so much increased as to permit the steam to exhaust at an early part of the stroke. To obviate this, we must put lap on the exhaust side of the valve. This has a bad effect in causing the exhaust valve to close too soon. This will be seen clearly in the illustration of the geometrical action of the slide valve, Fig. 16. Let A B equal



Geometrical Diagram of the Slide Valve by Messrs. Long and Buel.

FIG. 16.*

* The curve from a to P should be more, as that below, e to B.

the length of stroke of our engine drawn to any scale. We will give the valve an amount of lap on the steam side equal to half the breadth of the steam port. The travel of the valve will then be equal to three times the breadth of the steam port. On A B, as a diameter, describe a circle which will represent the path described by the centre of the crank pin, while the piston is travelling twice the distance A B. Divide this circle into any number of equal parts, and draw perpendiculars to A B from every point of division. We shall thus determine the position of the piston corresponding to that of the crank at various points. With the same centre *t*, as that of the circle A *r* B *s*, describe a circle C *o* D *p* having the travel of the valve for its diameter. This will represent the path of the centre of the eccentric, during a revolution of the crank. When the crank is on the centre, the line connecting the centre of the crank pin and the centre of the shaft will be A *t*; so that if the valve had neither lap nor lead, the line connecting the centre of the eccentric and the centre of the shaft should take the direction *t r*, perpendicular to A *t*. But in the present case, when we have both lap and lead, we make *t u* equal to the sum of the lap and lead, and through *u*, draw a line parallel to *t r*.

Connect the point o where this line cuts the circle with the centre, and $o\,t$ will be the proper position for the line connecting the centre of the eccentric and the centre of the shaft, when the crank is on one centre. When the crank is on the other centre, this line will appear at t F. Divide the circle C o D p into the same number of equal parts as we divided the circle A r B s, and draw perpendiculars to $o\,p$ from every point of division. The lengths of these perpendiculars show the distances travelled by the valve at various points. Now, let A B represent the centre of the exhaust port. Then draw $a\,b$ and $c\,d$ to represent the width of one steam port, and $e\,f$ and $g\,h$ for the other. Make $a\,i$ equal to L W, the steam lead, and draw a line $i\,k$ parallel to $a\,b$. This is the line to which all our measurements must be referred, since the valve commences to move from this position. Thus, when the crank has moved the distance A 1, the centre of the eccentric has moved the distance o 1, and the perpendicular distance of this point 1 from $o\,p$ will be the distance the valve has travelled. Lay off this distance below the line $i\,k$ on the first perpendicular, and the point so determined will represent one position of the valve. Similarly, when the crank has travelled the distance A 2, the centre of the

eccentric has passed over $o\ 2$, and the valve has travelled the perpendicular distance between 2 and $o\ p$. Lay off this distance on the second perpendicular below $i\ k$, and we determine the position of the valve at another point of the stroke. Find the position of the valve in this manner at every point of division, and through the points so found draw a curve which will represent all the positions of the valve during one revolution of the crank. It must be observed, in laying off perpendicular distances from $o\ p$, that all points to the right of $o\ p$ are positive, and are laid off below $i\ k$, while all points to the left of $o\ p$ are negative, and must be laid off above $i\ k$. We have not yet determined the amount of exhaust lap, but this can readily be fixed, now that the motion of the valve is represented. When the piston has made one stroke, the crank is at B and the valve is at F. Now, if the face of the valve was only $1\frac{1}{2}$ times the width of the port, the whole port would be open for the steam to exhaust. So we must put lap on the exhaust side of the valve, and we put on enough to have the exhaust lead equal to H I. This gives us the width W E of the valve face.

From the curve $a\ i$ P, we can readily find the position of the valve, corresponding to any posi-

tion of the piston. Thus, when the piston has travelled the distance A q, the valve is at v, and the distance the port is open is equal to the perpendicular distance between v and the top of the port $a\,b$. At x, where the curve cuts $a\,b$, the port is closed, and the steam expands during the remainder of the stroke from y to B.

We can readily lay down the curve described by the lower extremity of the valve. The distance $e\,m$ between $e\,f$, the top of the port, and $m\,n$, is the exhaust lead, and we have only to transfer the distances of the various points in the first curve from $i\,k$ to their respective distances from $m\,n$, on the same perpendicular on which they were first laid down. By this curve we see that the exhaust opening is closed at j, when the piston has travelled the distance A z.

The Geometry of the Slide Valve, by N. P. Burgh.

The first consideration in relation to this subject is the principles of the requirements which form the action of the valve during the operations of lead, full steam, and cut off.

The term "lead" is the position of the valve to allow a certain amount of steam to enter the cylinder before the piston completes its stroke, so that the motion of the valve is said to be in ad-

vance of that of the piston on the return stroke of the latter.

The next term is "full steam," which is the position of the valve when it is at the full stroke but the piston continuing its motion in the same direction as before.

The third term, "cut off," is the position of the valve when it has returned and closed the supply port or opening, and the piston is advancing as before, but the valve moving in an opposite direction.

We have next to remember the fact that the slide valve receives its motion from an eccentric or small crank, and that in either case the two motions—of the valve and eccentric—are sliding and rotary.

Now, on pages 40 and 41 of this work, there is illustrated a diagram, Fig. 9, of the several angles of the cranks in relation to the respective positions of the piston or connecting-rod pin; and it is also shown that the advance of the crank pin does *not* agree with the relative position of the piston in any case *due to the length of the connecting-rod. Obviously, then, as the relation of the positions of the piston and crank pin is affected by the length of the connecting-rod, so will the position of the slide valve and the angle of the eccentric be affected by the length of the eccentric rod;* for both

the piston and valve *slide* on their travel, while the crank pin and the centre of the eccentric *revolve;* therefore the action is the same in each case.

To further explain these matters and their relation to the outside lap of the slide valve, we will follow the motion of the crank pin and centre of the eccentric from the commencement of the stroke of the piston to the point of the cut off of the steam. Assume, therefore, that we have an engine before us, the motion of which we direct attention to. When the crank is horizontal the eccentric is at the opposite side of the centre of the shaft, at an angle *above* the horizontal centre line—due to the lead required—the crank's motion being upwards.

We may here mention that the *position* of the eccentric in relation to the crank is always at an angle towards the direction of the crank's motion when the eccentric rod is connected directly to the side valve rod pin.

The valve having admitted steam into the cylinder, the piston is then impelled back, when the crank moved, and, as the eccentric is keyed on the shaft at the angle alluded to, its position in relation to the crank is fixed, without alteration. The valve is now being pushed back until the eccentric is horizontal, which gives the

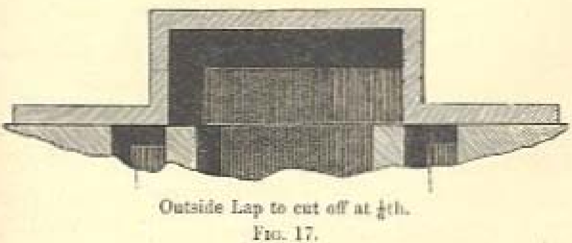
full area for the admittance of the steam—hence the term "full steam."

We may add that the motions of the crank and eccentric are, of course, in opposite directions, for when the crank rose above the horizontal line, the eccentric moved down towards it.

The crank pin continuing to move in its path and the eccentric also, the valve is pulled back, and thus the port is closed; at this point the "cut off" ensued. Then the angle of the crank and the length of its connecting-rod determined the position of the piston at the commencement of expansion.

It is apparent, therefore, that the outside lap of the valve is due to the grade of expansion, or the angle of the crank when that operation ensues; and, as an illustration of the fact, the following diagrams are introduced which relate especially to the diagram, Fig. 9, in pages 40 and 41:

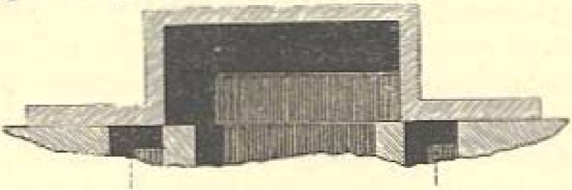
The valve, Fig. 17, is intended to cut off at



Outside Lap to cut off at $\frac{7}{8}$th.

FIG. 17.

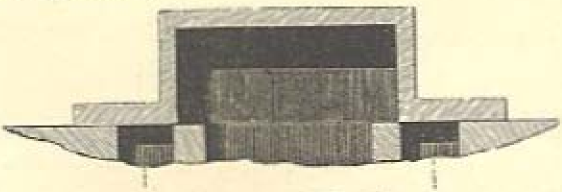
one-sixth of the stroke of the piston, or when the angle of the crank is at the point 1—see the diagram Fig. 9.



Outside Lap to cut off at $\frac{5}{8}$ths.

FIG. 18.

The vertical dotted lines in Fig. 17—as in the remainder—refer to the width of the opening caused by the valve, hence, in all cases, the half travel can be understood. The degree of expansion first given is perhaps rather higher than the general practice, hence the area of the cylinder is increased, the steam opening being reduced in width, but not in area, the length making good the requisition.



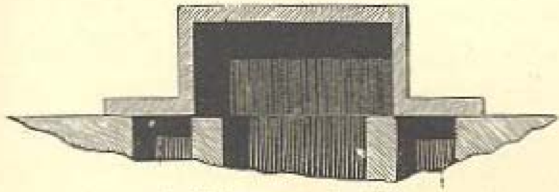
Outside Lap to cut off at $\frac{3}{4}$ths.

FIG. 19.

Fig. 18 represents a valve and ports to cut off

at one-third; in referring to the diagram, Fig. 9, the point 2 defines the angle of the crank. The lap in this case, also the opening, are both varied in relation to that for Fig. 17.

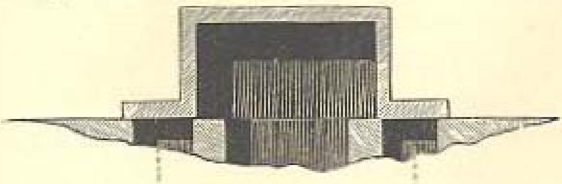
The next, Fig. 19, illustrates a valve and ports arranged to cut off at half-stroke, which is a good proportion, giving practical results—refer to the diagram, Fig. 9, again, where the point 3 denotes the angle of the crank.



Outside Lap to cut off at $\frac{2}{3}$ths.

FIG. 20.

Fig. 20 is a valve and ports to cut off at two-thirds of the stroke. The angle of the crank can be seen, on noting the position of the point 4 in the diagram in connection.



Outside Lap to cut off at $\frac{3}{4}$ths.

FIG. 21.

Fig. 21 is introduced merely to illustrate the greatest extent the piston should advance before the steam is cut off. The steam supply openings in all these examples are alike, hence the proportion of lap can be readily understood. In this last figure the valve cuts off at five-sixths of the stroke of the piston; the angle of the crank can be seen on referring to the diagram.

It may be mentioned again that the lead of the slide also regulates the travel of the valve, also that the angle for the eccentrics is determined from or by the advance of the valve on the piston. The advance of the valve from its full stroke will therefore be equal to the width of the opening, minus lead; for example, let it be assumed that the opening caused by a valve equals one inch, and the lead one-eighth of an inch, then seven-eighths of an inch *must be the advance of the valve from its full stroke when the piston is at the extremity of its path.*

The difference required in the outside laps of valves is more practically determined by diagrams than by calculation; due, of course, to the angles and versed sines. This axiom should always be remembered, viz., *that the path of the centre of the crank pin is virtually that of the centre of the eccentric.* Thus, the arc passed through by

the crank pin, at a given radius, must be that of the centre of the eccentric at the same radius, irrespective of the angles or relative positions of either.

To put this matter in the most practical form, we will again refer to the motions of the eccentric and crank, and their relative positions during "lead, cut off, and full supply." The crank on leaving the horizontal position started the eccentric from its angle, when the lead terminated and both moved on their paths, the valve allowing the supply of steam until the action of cut-off occurred. It is evident, then, that the crank during this operation moved through a certain length of arc, and if the limits of that travel were joined by a line—thus forming a chord of the travel—the versed sine will belong to the crank circle, but relates also to the several positions of the eccentric and valve at the points of "lead, full supply, and cut off."

Obviously also as the eccentric is keyed on the shaft, if its position for lead and its position for cut off are joined by a chord, the versed sine of the arc of that travel will be the versed sine of the eccentric circle.

To illustrate the full meaning of this latter conclusion, we have condensed the term and

named it the "versed sine of the eccentric," to which it actually belongs, and have introduced the diagram Fig. 22 as an explanation.

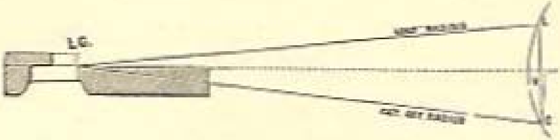
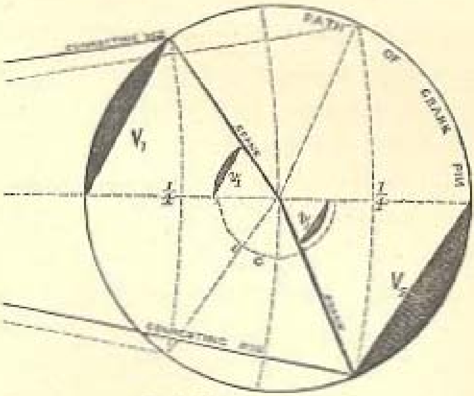


Diagram of Burgh's Mode of producing the "Versed Sine" of the Eccentric.

FIG. 22.

This represents a steam port and end of the lap portion of the valve at full travel; the dotted line L is the position of the valve for the "lead," and at C when it "cuts off." The lines L L and C C are the length of the distance between the centres of the crank shaft and block pin in the link, the centre of the pin being virtually the end of the slide valve. Then on the point L at the dotted line, inside the port, as a centre describe the arc cutting the plane line and the eccentric circle at L, and with C, also inside the port, as a centre describe an arc cutting the circle at C; the distance between the points of intersection on the plane line is the amount of the lead, or as the space between the dotted lines L C inside the port. Next join L C on the circle by a chord or a straight line, and the width at V is the *versed sine of the eccentric.*

As the arc of the circle of the eccentric's travel has been drawn in the preceding diagram, to prove the veracity of the "versed sine of the eccentric," it is necessary to explain how to obtain the radius of the arc *first*, and *after that* the remaining proportions. To do this practically we must fall back on the utility of the axiom "that the path of the centre of the crank pin is virtually that of the centre of the eccentric." We therefore next direct attention to the diagram Fig. 23, which relates to a width of



Scale 1 inch = 1 foot.

Burgh's Mode of determining the Lap of the Slide Valve, Points of Cut-off, and Positions of the Eccentrics.

Fig. 23.

steam opening caused by the slide valve equal to

$\frac{3}{4}$ in. and the lead $\frac{1}{4}$ in., so that when the piston was at the end of its stroke the slide valve had uncovered the port $\frac{1}{4}$ in., and therefore there was $\frac{1}{2}$ in. more travel required to complete the stroke of the valve. The circle of the path of the crank pin is 2 ft. in diameter, and the length of the connecting-rod 5 ft. The grade of expansion is $\frac{1}{4}$th, to which is due the relative position of the angles of the crank, in connection also with the length of the connecting-rod.

Having, then, proceeded thus far with the explanation of the construction, we will imagine now that the remainder of the diagram *has to be filled in*, and thus explain how to do it.

The chords V_1 and V_2 are drawn to depict the "versed sines of the crank." It will be noticed that the length of V_1 is shorter than V_2, although the lineal advance of the crank pin is equal on each side, and that the versed sines bear the same relation. Now, as the present positions of the cranks bear a strict relation to the entire stroke of the piston, therefore the versed sines bear a relation to the lineal advance of the piston or crank pin. Then as the crank pin has moved lineally 6 in., the versed sine of the arc joined by the chord V_1 is $1\frac{1}{2}$ in.; therefore as the grade of expansion is $\frac{1}{4}$th of the whole stroke, or the dia-

meter of the crank circle, the versed sine is $\frac{1}{4}$th of the advance of the piston, or $\frac{1}{16}$th of the diameter of the circle.

Next as to the versed sine of the arc joined by the chord V_2. In this case it is 2 in., or $\frac{1}{3}$rd of the lineal advance of the crank pin, and therefore only $\frac{1}{12}$th of the diameter of the circle; then assuming that V_1 and V_2 refer to the versed sines also, $V_1 = 6 \div 1{\cdot}5 = 4$, and $V_2 = 6 \div 2 = 3$; next, $V_1 = 24 \div 1{\cdot}5 = 16$, and $V_2 = 24 \div 2 = 12$, so that the mean sum of the versed sines in relation to the radius of the crank circle are known by $12 + 16 = 28 \div 4 = 7$, which we shall again refer to further on.

Having settled the meaning and proportions of the "versed sines of the crank," we have now to remember what the valve must have been doing during the movement of the crank pin, and that the paths of the eccentric's centre and crank pin are virtually the same. When the crank was horizontal the "lead" was given, and the angle of the eccentric, as shown by Fig. 22 at L, in page 64; and when the crank pin rose to the point on the circle due to its lineal advance 6 in., the valve closed the port or "cut off" the steam, and the eccentric was at *c*; therefore as the eccentric is keyed on the shaft, at whatever distance the

crank pin moved on the circle, the eccentric moved *virtually* through the *same length* of arc: then, as the versed sine of the crank bears a strict relation to the grade of expansion, so will the versed sine of the eccentric strictly refer to the outside lap of the valve, which is due also to the grade of expansion; for example, as the valve, has uncovered the port equal to the lead when the crank is horizontal, it completes its stroke when the eccentric is horizontal, and closes the port or cuts off the steam when the centre of the eccentric has moved lineally from its horizontal position, equal to the width of the opening caused by the valve for full steam.

It is obvious, then, that in Fig. 22, page 64, the "lead" is omitted in the "versed sine of the eccentric," because the valve has formed the lead when the crank was horizontal, and it is from that position that the chord of the arc of supply steam starts, and then intersects with the circle at the point of cut-off, from which arc the versed sine of the crank is known. Therefore, as the versed sine of the crank *omits the lead* in its dimension, and the paths of the crank pin and centre of the eccentric are virtually the same, the versed sine of the eccentric *must* omit the lead also; hence this formula, *the versed sine of the*

eccentric = width of steam supply opening caused by the slide valve, minus the lead, but with very short eccentric rods *half* the lead.

We will now return to Fig. 23, in page 65, and determine how to produce the arcs v_1 and v_2, which refer to the travel of the valve. Now, the versed V_1 is $1\frac{1}{2}$ in., or equal to $\frac{1}{8}$th of the radius of the crank circle; then, as the versed sine of the eccentric is $\frac{1}{2}$ in., because the width of opening is $\frac{3}{4}$ in., and the lead $\frac{1}{4}$ in.; from $\frac{3}{4}-\frac{1}{4}=\frac{1}{2}$ in. Now, the angle of the crank in relation to its travel is also the angle of the eccentric from its starting point, which proves also that the versed sines of the crank and eccentric refer to each other proportionately, inasmuch that if the versed sine V_1 is $\frac{1}{8}$th of the radius of its arc, the versed sine v_1 will also be $\frac{1}{8}$th of the radius of its arc, which is $\frac{1}{2}$ in., and results that $\cdot 5 \times 8 = 4$ in., the radius of the arc v_1. Referring next to the arc v_2, the versed sine V_2 is $\frac{1}{6}$th of the radius of that arc, then $\cdot 5 \times 6 = 3$ in. the radius of the arc v_2, the versed sine being $\frac{1}{2}$ in. also, as v_1. The following will be the formula: To find the radius of the eccentric circle for a slide valve with direct connection from the eccentric = [*radius of the crank circle* $\div$ *versed sine of the crank arc*] $\times$ *versed sine of the eccentric arc*.

It will be noticed that the versed sines of the two eccentric arcs are alike in the diagram, Fig. 23, page 65, but the radii are unequal, and that there is a dotted semicircle, whose radius is a mean sum of the two other radii. Now, should this mean radius be required to suit any purpose, it is obtained from the mean proportions of the versed sines of the crank, which is $\frac{1}{14}$th in this case, then $14 \div 2 \times \cdot 5 = 3 \cdot 5$ in., the radius of the dotted semicircle. Having the travel of the valve, we have also the throw of the eccentric; therefore the outside lap must be derived as shown by Fig. 10, in page 34, which = radius of eccentric circle − width of steam supply opening caused by the slide valve = outside lap of the valve. But if it is required to obtain the outside lap direct from the versed sine of the crank—which it really originates from, the formula will be thus: Outside lap of the slide valve = *divide the radius of the crank circle by the versed sine of the crank, multiply the quotient by the versed sine of the eccentric; the product, minus the width of supply opening caused by the slide valve, equals the outside lap of the slide valve.*

For example, in connection with the diagram, Fig. 23, page 65:

	Inches.
Width of opening caused by the slide valve	$\frac{3}{4}$
Lead	$\frac{1}{4}$
Versed sine of eccentric	$\frac{1}{2}$
Versed sine of crank (first)	$1\frac{1}{2}$
Versed sine of crank (second)	2
Diameter of crank circle	24

Then, $24 \div 1\frac{1}{2} = 16$, and $24 \div 2 = 12$, also $16 + 12 = 28 \div 2 = 14 \div 2 = 7$, and $7 \times \cdot 5 = 3\frac{1}{2}$; then $3\frac{1}{2} - \frac{3}{4} = 2\frac{3}{4}$ in., the outside lap for the mean travel of the valve. If V_1 and v_1 is used, it is arranged thus: $12 \div 1\cdot 5 = 8 \times \cdot 5 = 4$, then $4 - \frac{3}{4} = 3\frac{1}{4}$, the outside lap; but if V_2 and v_2 are preferred, then $12 \div 2 = 6 \times \cdot 5 = 3$ and $3 - \frac{3}{4} = 2\frac{1}{4}$, the outside lap, and the mean of $2\frac{1}{4} + 3\frac{1}{4} = 2\frac{3}{4}$ being the same as produced before.

CHAPTER V.

ON THE APPLICATION OF THE SLIDE VALVE AS AN EXPANSION VALVE.

To correctly define this question it is necessary that the action of the steam in the cylinder shall be first considered, and also the *time* for lead, cut-off, and expansion. Suppose the piston of a steam engine is at the complete stroke, and the crank on the dead centre, the steam is then acting on the piston, due to the lead of the valve, and the moment the crank moves the piston will be impelled, for the steam will then act with force due to its admittance; so that it is not erroneous to assume that the greatest effect of the steam on the piston is, when the valve is at full stroke, or the port widest open for the supply. The piston, be it remembered, is now supposed to be exerting its utmost efforts; due, of course, to the action of the steam. Presume the valve to return

and close the port at a given point of the stroke of the piston. At this point expansion commences, the piston being propelled after that by the elasticity of the steam. The time the steam is admitted into the cylinder is, while the valve is moving to and fro, or opening and closing the steam opening. The time the steam commences its actual propulsive power on the piston to that of its cut-off = width of opening × 2—lead. The valve, when last alluded to, was presumed to be at the edge of the port, expansion therefore being in full operation. Before the steam can exhaust, the valve must open the same port, but contrary in its motion.

The time occurring between the point of cutting off the supply and the opening the port for the exhaust is, of course, due to the outside and inside laps. Therefore the time allowed for expansion = outside lap + inside lap. This will be better understood by presuming the valve to have no laps, it will then be obvious that if additional length be added to the valve, that increase must be observed.

The time allowed for exhaustion is due to the speed or travel of the valve from a point to a point, or in principle as before stated for supply and expansion. Now, presuming the valve has

covered the port, and travelled so that the inner part is at the inside edge of the port, it can readily be understood that exhaustion must ensue instanter with the action of the opening. The valve now moves forward and backward for a given length, occupying thereby a certain time. If there was no lap inside, it is obvious that when the valve was at half stroke the time for exhaustion would be while the valve was opening and closing the port or opening, on the opposite end of the cylinder, outside lap being also considered. When there is an inside lap, the valve from the half stroke must move that sum or distance before the exhaustion commences; the valve then moves back due to the distance as before stated, and forward the same, less the inside lap. The formula for time of exhaustion, therefore, will be thus, outside lap + width of opening caused by valve × 2.—[inside lap × 2.]

By these formulæ—time being considered—an almost perfect knowledge of the action of the steam can be attained, its elastic force and the volume consumed at each stroke of the piston known; but as there may be, perhaps, those who cannot understand the application of the rules now noticed, also their practical use; a little further explanation will not be out of place. For

example, presume an engine of 50-horse power nominal; length of stroke, 2 ft.; diameter of cylinder, $47\frac{7}{8}$ in.; utmost grade of expansion, $\frac{1}{6}$th of the stroke. Let it be assumed the travel of the valve to be $9\frac{7}{8}$ in. unalterable. The cubic contents of the cylinder will be deduced in the usual form $1800 \times 24 = 43200$ cubic contents in inches. The space occupied by the steam at $\frac{1}{6}\text{th} = \frac{43200}{6}$—$7200$, or $1800 \times 4 = 7200$. This, then, shows that the expansive powers of the steam must emanate from a volume of steam whose capacity equals $\frac{1}{6}$th of the total contents of the cylinder. Now the amount of expansion is, of course, due to the capacity occupied after the termination of the *entrée* of the steam; to define, therefore, the amount of expansion, time and space must be duly considered. The piston in the present example is presumed to have moved 4 in., or $\frac{1}{6}$th of the stroke. The valve has closed the port, and has to travel the outside lap + inside lap, until the steam is released from the cylinder. To further demonstrate the present theory, time must be noticed. Let it be assumed that the engine, or rather the piston, moves at the rate of 160 strokes per minute—two strokes, of course, making a revolution. The

valve, be it remembered, moves at the same speed, action only being now alluded to. Now, when the piston is at the extremity of its stroke, the valve has performed the greater portion of its movement to the utmost point, and, consequently, in a given time moves in a contrary direction. The relative positions of the valve to the piston are due to the arcs passed through rotatively, each being the same in proportion. Circumferences of circles bear an equal relation to each other as their diameters; due, of course, to the constant number used. Thus, a circle $47\frac{7}{8}$ in. diameter equals 150·4 in. in circumference; a circle $9\frac{7}{8}$ in. in diameter equals 31·02 in. in circumference. The proportion of the diameter will be as 1 is to 4·84; then $\frac{150{\cdot}4}{31{\cdot}02}$—4·84, proving thereby that the arcs passed through are the same in proportion lineally. To produce the amount of expansion by calculation, in the absence of diagrams, is a difficult matter perhaps, without accomplishing a correct result, due to the length of the connecting and eccentric rods, the versed sines of which bear strict reference to the action of the valve. It is, therefore, advisable to practically set out all the points of contact and requisition, in order to produce a

perfect result. For example, when the piston is at full stroke, the eccentric is at a given angle, due to the width of opening or port and the lead determined or decided on. Now, as before stated, the crank moves through a certain arc, and the *eccentric virtually through the same*, or proportionately the same; but the distance on the plane line is entirely different; due, of course, to the angle of the chords. In the case of the crank the chord is at an angle, but that for the valve may be termed almost perpendicular. It is certain that calculations alone cannot produce a correct result in this matter, therefore diagrams are required. And we may add, that as the use of steam expansively produces economy, the lap and lead of the valve, in proportion to the stroke of the engine, govern expansion.

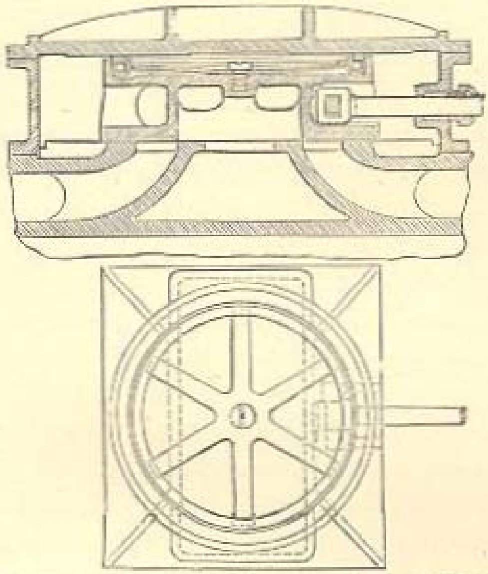
CHAPTER VI.

ILLUSTRATIONS AND PROPORTIONS OF MODERN SLIDE VALVES IN ACTUAL PRACTICE.

The examples of valves that are illustrated in this chapter are, for the greater part, by the most eminent firms in England and Scotland. The leading dimensions of each are given, so that they form a practical guide for other examples, of similar types, that may be required.

Fig. 24 represents the sectional elevation of a single ported valve and casing by Messrs. Ravenhill and Hodgson, with the ports and a portion of the passages in the cylinder for a marine oscillating engine of 125-horse power nominal, being one of a pair for the same cylinder, situated on each side fore and aft.

The main features are that the valve is flanged at the back, with a circular rim, sufficiently deep to contain three rings and a spring within it,



Messrs. Ravenhill and Hodgson's modern single ported Slide Valve.

FIG. 24.

The spring has six branches—shown in the plan—and is sustained, centrally, by a stud. The ring against which the spring is acting, or in contact with, is right-angular in section, with a circular projection at the back to receive the extremities of the branches of the spring; this ring is termed the spring ring. The next ring is the fitting ring, which is merely a plain circular piece of metal turned to fit the bored

rim portion at the back of the valve. The third is the face ring, which is in contact with the cover of the valve casing. This ring is the main one, as on it depends whether the steam enters the recess within the rim.

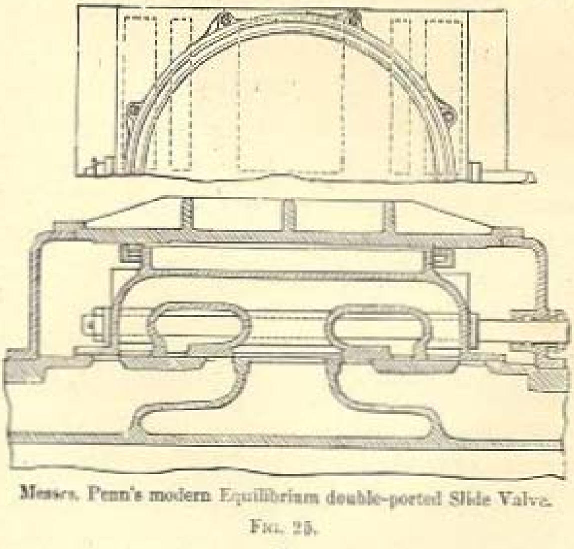
It will be noticed from the plan, that although the diameter of the face ring is almost equal to the length of the valve, it is not equal to the width; therefore a certain amount of valve face flange surface is exposed to the action of the steam, and shows the valve is not entirely equilibrium. Now if it is required to make a valve of this class equilibrium, the back flange of the recess portion must be nearly the same size and shape as the face flange of the valve, so that the steam has equal surfaces to act against in either direction.

The following are the mean proportions of this valve, Fig. 24:

	Inches.
Width of each supply port in cylinder	$3\frac{1}{2}$
Width of exhaust port	$5\frac{3}{4}$
Width of bars	$1\frac{7}{8}$
Width of opening caused by the valve	$1\frac{1}{2}$
Length of ports.	$20\frac{5}{8}$
Outside lap of the valve	$2\frac{3}{4}$

	Inches.
Stroke of lap of the valve	$8\frac{1}{2}$
Inside lap of do.	none.
Thickness of face flange of do. . . .	$\frac{7}{8}$
Length of do. do.	22
Width of do. do.	24
Depth from valve facing to ring facing on cover	$6\frac{5}{8}$
Diameter of valve rod	$1\frac{3}{8}$
Centre of rod from valve facing . .	$2\frac{1}{4}$

The valve that we next direct attention to is that illustrated by Fig. 25, for a marine engine,



Messrs. Penn's modern Equilibrium double-ported Slide Valve.

FIG. 25.

double trunk, of 250 horse power nominal, constructed by Messrs. John Penn and Sons. This valve is the equilibrium double ported kind, and is fitted with a packing ring at the back with adjusting studs, click ratchets, and springs, which are illustrated and described in detail in page 102. The form of the ring is circular, as shown by the plan, from which also the number of the studs can be known. In the sectional elevation, the sections of both the valve and casing are shown, also the ports and a portion of the passages in the cylinder.

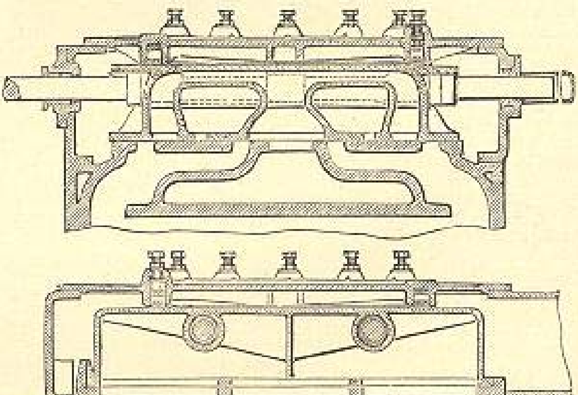
The slide rod passes through the valve, it will be seen, instead of being secured by a pin at the front end, as in Fig. 24; by this method a steadier motion for the valve is said to be ensured than with the other, because as the rod passes through the valve, and is secured to it by a nut and collar, the rigidity of the rod is conveyed to the valve by the connection. Of course the size and weight of the valve greatly determine this difference of the mode of connection; inasmuch that, what is sufficient for a valve of certain dimensions and weight, may be of no use in practice for a valve of larger proportions under similar circumstances of position.

The leading dimensions of the valve, Fig 25, are :

	Inches.
Width of each supply port in the cylinder	4
Width of exhaust port do.	$10\frac{7}{8}$
Width of narrow bars	$2\frac{11}{16}$
Width of large bars	$13\frac{1}{16}$
Width of opening caused by the valve	$2\frac{1}{4}$
Length of ports	$45\frac{1}{2}$
Outside lap of the valve	$2\frac{3}{4}$
Stroke of the valve	10
Inside lap of the valve	$\frac{1}{32}$
Width of supply port in the valve	$3\frac{1}{8}$
Thickness of the flange	$1\frac{3}{8}$
Length of do.	$64\frac{3}{8}$
Width of do.	50
Depth from valve facing to the face of packing ring	17
Outer diameter of packing ring	$47\frac{1}{2}$
Width of do.	2
Diameter of valve rod in valve	$2\frac{3}{4}$
Centre of rod from valve facing	$4\frac{1}{2}$

Within the last two or three years it has been the practice of many engineers to use two rods in the place of one for the slide valve, so that a better guidance is obtained, and therefore a more certain action results. The benefit of this is, of course, obvious; for if a valve slides unsteadily

on its seat, or wears harder on one side than the other, the first evil produces unequal action, while the latter allows leakages to occur, and thus the duty of the valve is greatly impaired Another reason for the use of two rods is that when a valve is wide in proportion to its length the power to move it should not be concentrated at the centre, but rather be distributed as much as practicable. As an illustration of this, the Fig. 26 is introduced, this being an example by Messrs. Napier.



Messrs. Napier's modern Equilibrium double-ported Slide Valve with two Rods.

FIG. 26.

This illustration depicts the valve and casing in longitudinal and transverse sections; the steam ports and passages of the cylinder are also shown, so that the relation is easily understood. The mode of packing the valve facing at the back is entirely reverse to that in Fig. 25; in that case the ring is recessed in the *valve*, but in this case it is inserted in the *cover*. The adjustment is accomplished by set studs and spiral springs above the two packing rings, which greatly assist the elasticity of the gasket or india-rubber. A detail of this is shown in page 103.

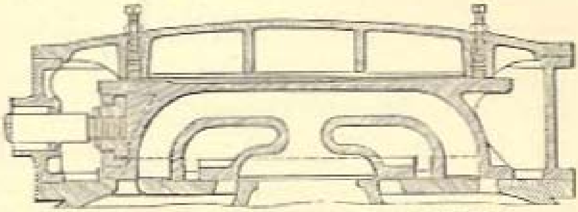
The valve rods pass through the valve, are secured by nuts, and prolonged to guides secured to the back end of the casing; it will be noticed also that the position of the rods are higher from the cylinder facing in this case than for the previous example, Fig. 25. The transverse view shows the steam passages and the section of the rods and bosses, also the packing rings and adjusting studs. The main dimensions of this valve are:

	Inches.
Width of each supply port in the cylinder	$3\frac{1}{4}$
With of exhaust port do.	9

	Inches.
Width of narrow bars in the cylinder .	$2\frac{1}{2}$
Width of large bars do.	$13\frac{3}{8}$
Width of opening caused by the valve.	$1\frac{3}{4}$
Length of ports	63
Outside lap of the valve	$3\frac{3}{4}$
Stroke of the valve	11
Inside lap of the valve	$\frac{1}{8}$
Width of supply port in the valve . .	$2\frac{3}{8}$
Thickness cf the flange.	$1\frac{3}{8}$
Length of do.	$61\frac{3}{4}$
Width of do.	72
Depth from valve facing to the packing ring	$12\frac{1}{2}$
Outer diameter of packing ring . . .	49
Width of diameter of do.	$2\frac{1}{4}$
Diameter of valve rod in valve . . .	4
Distance between centres of valve rod .	30
Centre of rod from valve facing . . .	$8\frac{1}{4}$

Another example of equilibrium valve is shown by Fig. 27, constructed by Messrs. Rennie for one of a pair for engines of 350 horse power collectively nominal. The greatest peculiarity in this example is the mode for securing the rod, which is a loose T nut and a narrow lock nut outside; the former is slid in laterally, and the rod retains it in position.

The packing ring is almost similarly arranged as in the previous example by Messrs. Napier, the only difference being that in the present case the spiral springs are not used. The following are the leading proportions of this valve. Fig. 27.



Messrs. Rennie's Equilibrium double ported Slide Valve.

FIG. 27.

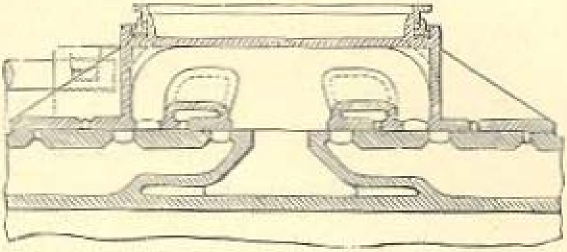
	Inches.
Width of each supply port in the cylinder	$2\frac{1}{4}$
Width of exhaust port in do. . . .	10
Width of narrow bars in do. . . .	2
Width of large bars in do.	$7\frac{1}{8}$
Width of opening caused by the valve	$1\frac{1}{2}$
Length of ports	50
Outside lap of the valve	$1\frac{5}{8}$
Stroke of the valve	$6\frac{1}{4}$
Inside lap of the valve	none
Width of supply port in the valve .	$1\frac{9}{16}$
Thickness of the flange	$1\frac{1}{4}$
Length of do.	42

	Inches.
Width of the flange	56
Depth from valve facing to packing ring	$9\frac{3}{4}$
Outer diameter of packing ring . .	36
Width of do.	$1\frac{1}{2}$
Diameter of the valve rod (screwed) .	$2\frac{1}{4}$
Centre of rod from valve facing . .	$5\frac{1}{4}$

Having sufficiently illustrated the double-ported valve, we will now proceed to the treble-ported kind, which is illustrated by Fig. 28, and has also been inserted on page 24 in connection with the formulæ for its proportions.

The design of this valve has no particular feature to render it superior to the examples preceding; for, as in Fig. 26, the exhaust steam passage over the supply openings are angular, to reduce the height of the valve, while the method for securing the rod is by a key, instead of a nut as in Fig. 27. The packing ring, however, for this valve is of a peculiar kind, and a section of it, at a large scale, is given in page 104.

The main purpose of the valve is, that it is at once a slide valve and an expansion valve, and has been constructed by Messrs. Maudslay, Sons, and Field, being one of the many that they have fitted to three-cylinder engines.



Messrs. Maudslay's treble-ported Equilibrium Slide Valve.
FIG. 28.

The example under notice is precisely similar to a valve the firm constructed for one cylinder —of three engines of 300 horse power collectively nominal. It receives its motion from a crank, and the position of the valve is opposite the connecting-rod of the main crank and engine shaft, the advantage of which is noticed in page 38, and illustrated and further described in page 42. As this valve is used as an expansion valve, it is proportioned to cut off from $\frac{1}{4}$th to $\frac{1}{8}$th of the stroke of the piston; therefore the outside laps are unequal, also the wide bars in the cylinder, to which we have referred in the table of dimensions following:

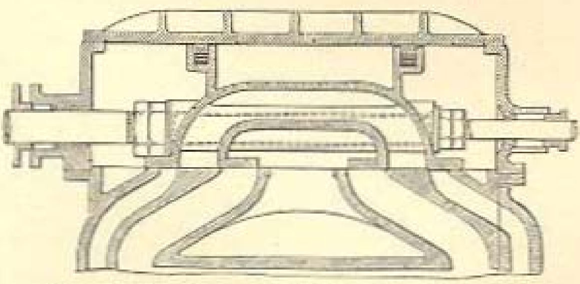
	Inches.
Width of each wide supply port in the cylinder	$1\frac{3}{4}$
Width of each narrow port in do. . .	$\frac{3}{4}$

	Inches
Width of exhaust port in the cylinder	$4\frac{1}{2}$
Width of narrow bars	$1\frac{5}{8}$
Width of large do.	$5\frac{3}{4}$
Width of largest do.	$6\frac{1}{4}$
Width of opening caused by the valve	$\frac{3}{4}$
Length of ports	46
Outside lap of the valve (mean) . .	$1\frac{3}{4}$
Stroke of the valve	5
Inside lap of do.	$\frac{1}{16}$
Width of the supply ports in the valve	$\frac{5}{8}$
Thickness of the flange	$\frac{7}{8}$
Length of do.	44
Width of do.	48
Depth of valve facing to packing ring	$9\frac{3}{4}$
Outer diameter of packing ring . . .	$26\frac{1}{2}$
Width of do.	1
Diameter of the valve rod	$2\frac{1}{4}$
Centre of rod from valve facing . .	3

All the valves that have for the present been noticed are for single cylinders. We, therefore, direct attention next to an arrangement of valve for *two* cylinders, commonly known as the "Compound" system. This is illustrated by Fig. 29, and has been constructed by Messrs. Dudgeon for engines with four cylinders of 350 horse power collectively nominal.

There is no novelty in this design, it being the ordinary form for the purpose. The arrangement of the packing ring at the back is the simplest of all the methods that have been alluded to, and has been largely adopted by several of the leading firms. Messrs. Penn, for example, have successfully used it for the slide valves of their engines of the largest power which they have constructed.

The simplicity lies in the fact that there are no adjusting studs, springs, or any mechanical appendages; it merely being a recess on the back of the valve, partially filled with india-rubber, and the face ring laid thereon as shown. The



Messrs. Dudgeon's Slide Valve for Compound Engines.

FIG. 23.

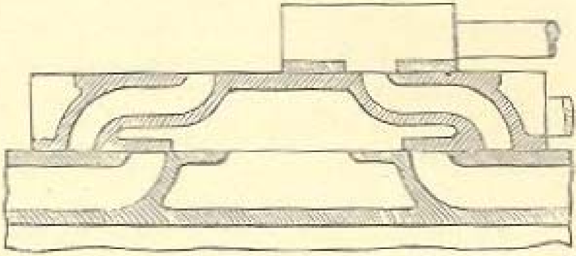
rod passes through both the valve and casing, so as to effectually guide the former.

The leading proportions of this example are as follows:

	Inches.
Width of the high pressure supply port in the cylinder	2
Width of low pressure port in do. .	4
Width of exhaust port in do. . . .	$6\frac{3}{4}$
Width of large bar in do.	$4\frac{1}{2}$
Width of narrow bar in do..	1
Width of opening caused by the valve	$1\frac{11}{16}$
Length of ports	30
Outside lap of valve	$1\frac{7}{16}$
Stroke of valve	$6\frac{1}{4}$
Inside lap of valve	$\frac{3}{16}$
Thickness of the flange of valve . .	1
Length of do.	$32\frac{5}{8}$
Width of do.	$33\frac{1}{2}$
Outer diameter of packing ring . .	23
Width of do.	$1\frac{1}{2}$
Depth from valve facing to packing ring	13
Diameter of valve rod	$3\frac{1}{4}$
Centre of rod from valve facing . .	$3\frac{5}{8}$

Following on from valves for compound engines, we next allude to slide valves for expansive purposes with one cylinder, generally known as "Expansion Slide Valves." An example of this type is illustrated by Fig. 30, as arranged by Mr. T. B. Winter, M.I.C.E. The cylinder

ports and passages are also shown, so that the relation of the whole detail is represented.



Mr. T. B. Winter's Expansion Slide Valves.

FIG. 30.

The main valve is of a peculiar section longitudinally, having passages through the body: the cut-off valve is merely as a flat plate with an opening or port centrally of its length.

The object sought after and attained with this arrangement is a sudden cut-off, and an extension of time for exhaustion in relation to single valves. Of course to produce this effect there are two eccentrics, separate for each valve, and a slotted connecting bar, screw and sliding block, to alter the grade of expansion as required. The grades accomplished with this arrangement are $\frac{2}{3}$rds, $\frac{1}{5}$th, and $\frac{1}{10}$th; and the main dimensions of the valves and ports for a pair of engines of 60 horse power collectively nominal are thus:

	Inches.
Width of supply port in the cylinder .	2
Width of bar in do.	3
Width of exhaust port	$6\frac{1}{2}$

Main Valve.

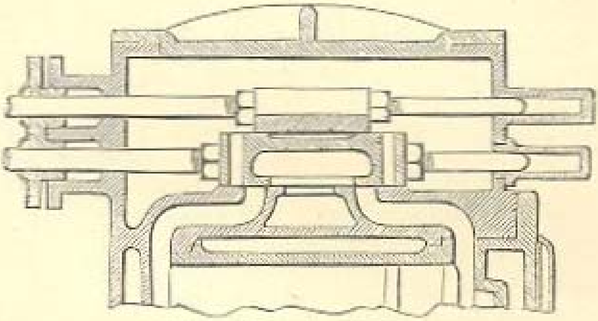
Width of supply port in the valve .	$1\frac{7}{16}$
Width of the outer bar in the valve front facing	$1\frac{5}{8}$
Width of the inner bar in the valve do.	4
Width of the outer bar in the valve back facing	6
Width of the central bar in the do. do.	$7\frac{1}{2}$

Cut-off Valve.

Width of port	$2\frac{7}{8}$
Length of valve	$8\frac{7}{8}$
Stroke of eccentrics	$6\frac{1}{2}$
Main slide cuts off at $\frac{2}{3}$rds.	
Cut-off valve from $\frac{1}{16}$th to $\frac{2}{3}$rds.	
Depth of main valve	4
Thickness of cut-off valve	$\frac{7}{8}$
Diameter of main valve rod	2
Centre from valve facing	$1\frac{3}{4}$
Diameter of cut-off valve rod . . .	2
Centre from valve facing	$1\frac{3}{4}$
Distance between centres of rods . .	4

A more popular arrangement for expansion

slide valves is illustrated by Fig. 31, which is more simple than that previously described. The main valve has two ports for supply straight through the body, while the cut-off valve is solid: the rods pass through the valves and casing, and thus a steady action is insured for the former. This arrangement is adapted plentifully in the North for all classes of engines.



Ordinary Expansion Slide Valves.

FIG. 31.

Mr. J. F. Spencer has fitted valves of this kind to marine engines often. In one instance, for a pair of inverted supplementary engines—cylinders 20 inches diameter—the following proportions were adopted:

	Inches.
Width of supply port in the cylinder .	$1\frac{1}{4}$
Width of bar in the cylinder . . .	$1\frac{1}{4}$

	Inches
Width of exhaust port in the cylinder	3
Length of port	13

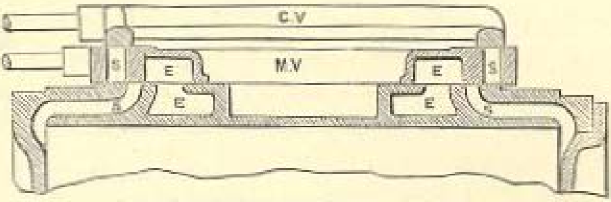
Main Valve.

Width of supply port in the valve .	$1\frac{1}{4}$
Width of outer bars in do.	$1\frac{1}{16}$
Width of inner do.	2
Width of exhaust space in do. . . .	$5\frac{3}{8}$
Depth of valve	$3\frac{3}{8}$
Width of do.	$15\frac{3}{4}$
Length of do.	14
Diameter of rod	$1\frac{1}{2}$
Centre from facing	$1\frac{1}{2}$

Cut-off Valve.

Thickness of valve	$\frac{3}{4}$
Length of do.	7
Width of do.	$14\frac{3}{4}$
Diameter of rod	$1\frac{1}{2}$
Centre from facing	$1\frac{1}{2}$
Stroke of the valves	4

For engines with a long stroke for the piston it is often the practice to arrange the ports as shown by the diagram Fig. 32, so that the supply steam passages shall be reduced in length as much as possible, and thus prevent an accumulation of steam in them during each stroke of the



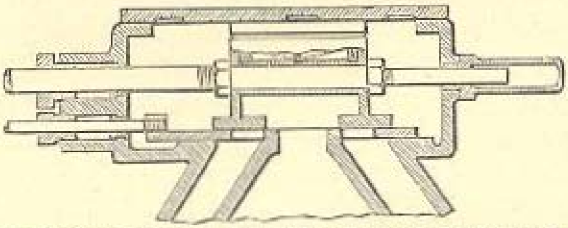
Ordinary elongated Expansion Valves.

FIG. 32.

piston. This arrangement for the ports will be understood by the letters of reference: C V is the cut-off valve, M V the main valve, S the supply ports and passage, and E the exhaust ports and passages; both the valves are at half stroke, therefore their proportions are apparent. Of course the stroke of the valve can be proportioned for any requirements; for if a long or maximum stroke is desired, one port at each end will be sufficient, but if half that stroke, with the same area, two ports are required; and if one-third of the stroke, three ports are necessary; so that it is really with expansive slide valves as with the ordinary kind, *i. e.*, the number of the ports regulates the stroke to a great extent. The matter, *in toto*, resolves itself into one simple fact, which is, that the *length of the time* for admission, cut-off, and exhaustion of the steam, is the real feature in the case, and is indeed the basis on

which the width of the supply opening and out side lap of the valve should be founded. This will be fully apparent when it is understood that the motion of the valve and steam piston are alike, and derive their motion from similar causes.

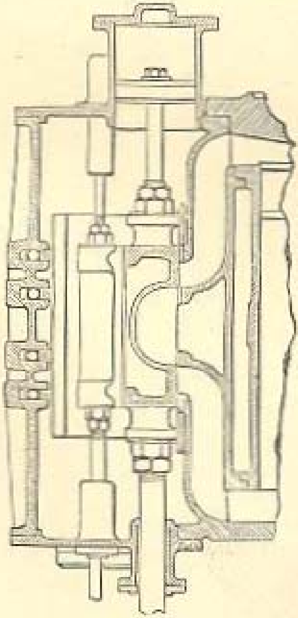
The most novel arrangement at present for valves of the class under notice is that by Messrs. Napier and Rankine, which is, that the main valve slides on a movable seat, as shown by Fig. 33.



Messrs. Napier and Rankine's Patent Slide Valve with the movable Seat.
Fig. 33.

To alter the grade of expansion, the seat moves forward and backward in the direction required, or similar to the shifting and motion for the ordinary cut-off valve. This arrangement is really that the cut-off valve is *under* the main valve instead of *over* it, and that the shifting and motion of the seating accomplishes the same effect as the top valve does when it is adjusted and put in motion.

To this point of description all the valves we have illustrated and referred to relate to horizontal engines. It sometimes occurs, however, that vertical inverted engines are preferred by some makers; we therefore direct attention to an ingenious arrangement of valves for this class of engine by Mr. J. F. Spencer, which he has adopted to relieve the link motion and disengaging gear from the weight of the valve; also



Mr Spencer's Mode of retaining Vertical Slide Valves in Equilibrium.

FIG. 34.

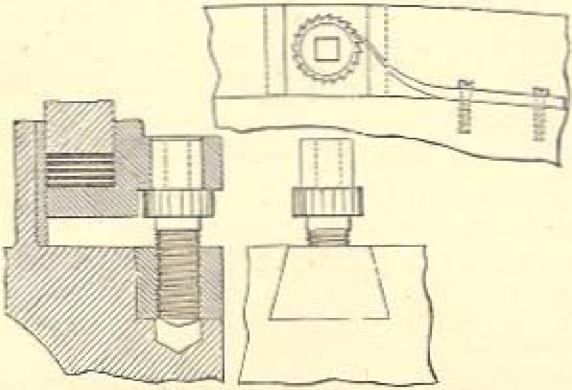
to keep it in position when the disconnection is required. The method is shown by the illustration Fig. 34. The main valve is held in any position by steam acting upon a piston which is secured to the rod above the valve. It is almost superfluous to state that the area of the piston is regulated by the pressure of the steam and weight of the valve.

CHAPTER VII.

DESCRIPTION OF PACKING RINGS FOR SLIDE VALVES.

THE most simple method for packing the back of the slide valve is by a recess cast around the body, and a ring of india-rubber inserted therein, with a ring of metal seated on it, as shown by the Fig. 29, in page 91. The form of the recess in plan can be round, square, or rectangular, to suit the valve or the extent of surface to be relieved from the pressure of the steam. This is, of course, self-adjusting, while the india-rubber retains its elasticity, which it does for a long time, even with high pressure steam.

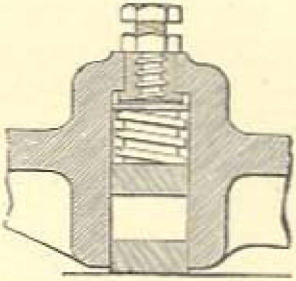
When adjustment is preferred, the arrangement to effect it, as illustrated by Fig. 35, is often used, which consists of a ring separated from the body, fitted with lugs, through which studs pass. Each stud has a ratchet pinion on it, to which a tongue spring is connected, to pre-



Modern adjustible Packing Ring for Slide Valves, by Messrs. Penn.
FIG. 35.

vent the stud from unscrewing: a brass nut is let into the body of the valve for the stud to fit into, and also to sustain non-corrosion.

The next mode of adjustment is shown by Fig. 36, which is a recess cast in the cover of the casing; two rings are inserted into it, with the gasket or india-rubber between them, the mechanical adjustment being attained by set-studs in the cover. To prevent the studs from sticking on their threads, bushes of gun-metal are preferred to the cast-iron cover for them to pass through. The elasticity of the packing is assisted by spiral springs enclosing the studs directly below each bush.



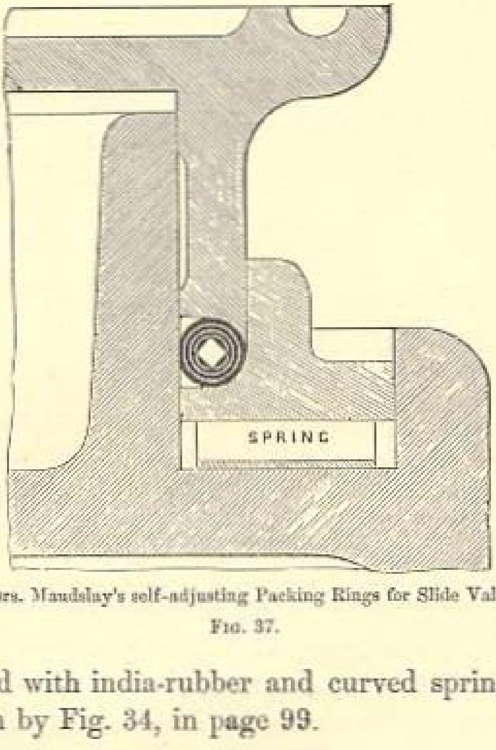
Set-Stud, Spiral Spring, and Sections of Rings for Packing Slide Valves, by Messrs. Napier.

Fig. 36.

The springs are sometimes omitted by some makers, and the set-studs only preferred, to adjust the rings, as shown by the sectional view of a valve casing and cylinder ports by Messrs. Rennie, in page 87, by Fig. 27.

Messrs Maudslay adopt the arrangement shown by Fig. 37, in page 104, which is a recess in the body, with an inner projection, the latter being encompassed by a ring of the peculiar section de picted. The metal ring in the recess contains an india-rubber ring of circular section, and is seated on curved springs; from this combination self-adjustment is attained, and the studs are therefore dispensed with.

It is preferred by some makers to use two separate rings recessed in the cover, packed



Messrs. Maudslay's self-adjusting Packing Rings for Slide Valves.

FIG. 37.

behind with india-rubber and curved springs, as shown by Fig. 34, in page 99.

CHAPTER VIII.

GENERAL OBSERVATIONS.

The preceding chapters have dealt with the dimensions to be attained, which are, of course, the primary considerations; but the arrangement of the ports on the cylinder facing has not been alluded to, therefore a few hints will not be out of place. Having determined the dimensions to be adhered to, the line of the valve facing should be first drawn,—a line at right angles will denote the centre of the exhaust port. Should a rib be required, half the width of the exhaust port will commence on each side of the same, a good proportion for which is, thickness of rib equals thickness of metal of passage, a fitting strip being provided on each side. Presuming the width of the exhaust port to be drawn, the inside or narrow bar must next be depicted, and from this the width of the supply port should be marked off. This completes the setting out of

the ports for the ordinary single ported slide valve. Should the equilibrium type be required, the same process of construction as before must be resorted to, and having thus far proceeded, the large bar is added, after which the width of the port supply. The raised face beyond the outer port, need not be much wider than the outside lap of the valve, for the reason that the least amount of surface in contact with the valve, the more perfect the connection, and the less the friction. Having thus far settled the main proportions of the ports and bars, set off the lap of the valve beyond the outer port, from this last point, the half travel plus the clearance, which will represent the commencement of the valve or steam casing. The width of the flange of the casing will depend on the diameter of the stud or bolt used, for which a good proportion is, half an inch to seven-eighths of an inch in diameter, and the pitch of the bolts eight times the diameter. The termination of the ports in the cylinder, is due to the length of the stroke and the depth of the piston, which will decide the distance between the ports, where entering the cylinder at each end. In some cases the piston is allowed to pass beyond or slightly cover the ports at each end of the stroke, so that the wearing surface may be equal throughout.

One important fact should be attended to, *i. e.*, the ports at the facings should have fitting strips on all sides. This matter is too often disregarded; it is generally overlooked that the fitting or shaping of the ports can be accomplished with greater facility, when strips are introduced than when the same are absent.

For oscillating engines, the setting out of the ports is much as that last described, but an additional supply port is introduced at right angles with those ordinarily arranged. This last mentioned port communicates with the steam passage leading from the supply trunnion, that for the exhaust being situated opposite.

In designing, or rather arranging, the flange for the valve casing, it should be remembered that the same level must be retained on all sides, both for the better means of planing and making a perfect joint. In setting out the steam passages, care must be taken to increase rather than decrease the dimensions, those for the ports being now alluded to. Should the area of the passage —by accident or carelessness of design—be less than that of the port on the face, the port is virtually reduced to the same size as the passage.

The proper means for ascertaining the depth of

the valve will be to consider the width of the supply port allowed for exhaustion. In some instances an increase is deemed necessary to allow a more free exhaust, but in practice the width of the port is the main consideration, as through it the exhaust steam must pass before it reaches the final or centre exhaust port.

The areas of the side steam openings, in double ported valves, must each equal half the area of one steam opening caused by the valve. By this proportion it will be understood that sufficient steam can enter the openings in the valve, on each side of the same. In some examples the area of these openings are reduced towards the centre, which may be said to be correct, as it allows a larger area for exhaustion above them. The area of the exhaust opening in the valve above the metal of the side openings should equal the area of one supply port in the cylinder. The longitudinal distance between the side supply openings in the valve, is due to the travel of the same, and width of one supply port in the cylinder, which must be duly considered to attain a correct action.

The steam passages of large cylinders are strengthened by ribs in a line with the cylinder, commencing at the valve facing and terminating with the passage itself. These ribs must be

equally disposed; should two be used, the length of the port should be divided into three equal portions; if one rib be required, two divisions will only be requisite. It is not imperative that the edges of the ribs should be on a level with the valve facing, but rather on a level with the lower edge of the fitting strip, which will be better both for planing and fitting. When the valve facing is at an increased distance from the cylinder, the ports should be arranged as before mentioned, but the passages as direct as possible, or at an angle rather than in a line with the body of the cylinder.

The position of the valve on the cylinder for horizontal engines will, of course, greatly affect the friction; for example, should the position be the top of the cylinder, the friction will be greatest; if at the bottom or the underside, the least; and if at the side of the cylinder, a happy medium will be maintained. This last mentioned position is most universal, due, of course, to the cause alluded to; and also as the best means of access for repair and renewal of the surfaces. Large valves have a guide at the bottom side, to prevent a lateral strain on the rod.

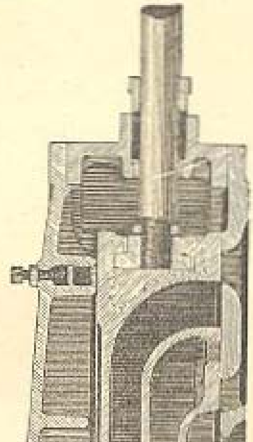
The modes of securing the rod in or to the valve are not much varied; in some cases the rod passes through the valve, and is secured at each

end of the same by nuts, and thus attains adjustment; also another mode is, to recess a nut in the front part of the valve, and screw the rod in the same; this is the best connection yet introduced for small valves. Two rods to each valve are now becoming more universal, but the mode of connection is sometimes the same as the last mentioned.

The stuffing box and gland for the rod should be properly considered; a good proportion for the diameter of the stuffing box is, diameter of rod $\times$ 1·5. Depth of stuffing box = diameter of the same $\times$ ·6 to ·5, while in some examples the depth will be found to equal the diameter. The depth of the gland is not of much importance, but a good proportion is attained by depth of stuffing box $\times$ ·75 to ·6. The diameter of the studs for adjusting the gland is not imperative, half an inch to three-quarters of an inch being often adopted, two studs only being used. The thickness of the flange of the gland generally equals the diameter of the bolt.

With reference to the position of the securing flange of the casing, some makers prefer an outside connection, while others adopt an inside flange. If external appearance be considered as worthy of notice, the latter is the better. The cover of the casing should be ribbed either in-

side or outside, the latter being the most simple, while the former presents the better exterior, as shown by Fig. 38.



Slide Valve. Casing and Rod, by Messrs. Rennie.
FIG. 38.

Having thus far duly considered the best mode of designing the slide valve, steam ports, and casing, allusion will now be made to the strains imposed by the steam. The casing, be it remembered, may be termed a box, within which a certain force is exerted, therefore, on the steam being admitted, each portion is operated on. The flanges are secured by the studs, hence the latter are subject to a tensile strain. The sides and ends of the casing are subject to a tensile and lateral or bulging strains, the former predominating. The cover and front portion of the casing are exposed to the greater effect of the pressure, due, of course, to the surface exposed, hence the cause for ribs, &c. The action of the steam on the valve is greatly counteracted by the curved form it generally assumes, hence the thickness of the metal may be considerably less than that for the casing. A good proportion is three-eighths to three-quarters of an inch for the thickness of the body part of the valve, and five-eighths to seven-eighths of an inch for the body part of the casing. The flanges in each case should be slightly thicker than the body part. The thickness of the ribs should be less than that of the body, as the former are not subject to the full pressure of the steam.

In designing the slide valve, whether the ordinary or equilibrium kind, strict attention should be given to the speed of the piston, pressure of the steam, grade of expansion, length of the connecting rod, lead of the valve on the piston, &c., matters which have been explained in detail. The only excuse for again reverting to these important facts is, to impress on the mind of the young Engineer that, of all the details of the steam engine, the most important is the "SLIDE VALVE"

INDEX.

A LIST OF THE

PRACTICAL SCIENTIFIC WORKS

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